STEAM POWER PLANT ENGINEERING

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PREFACE TO FOURTH EDITION

THIS work was first published in 1908. Many additions and changes were made in 1909 and in 1911. The development of Steam Power Plants has been so rapid during the past three years that it has become necessary to rewrite a considerable portion of the book. The chapters on Fuels and Combustion, Engines, Turbines, Condensers, Finance and Economics have been entirely rewritten and the others have been revised and enlarged. The chapter on Finance and Economics contains many operating charts showing current practice in power plant cost accounting, typical load curves illustrating daily, monthly and yearly load characteristics for a wide range in practice, and numerous tables giving recent results in cost of operation. Over three hundred new illustrations have been added and the entire text has been reset.

Although primarily intended as a textbook for engineering students it is hoped that the new edition will be of interest to practicing engineers.

G. F. G.

CHICAGO, ILL., *April*, 1913.

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STEAM POWER PLANT ENGINEERING.

CHAPTER I.

ELEMENTARY STEAM POWER PLANTS.

1. General. — The fundamental object of any power plant is the conversion of energy from one form into another at the least ultimate cost. This involves not only the cost of converting the energy into the desired form, but also the cost of distribution and application.

The most efficient plant, thermally, in the conversion of energy from one form to another, is not necessarily the most economical commercially, since the various items involved in effecting this conversion may more than offset the gain over a less efficient plant. There is no question as to the low cost of power generated by hydro-electric plants, but when the cost of transmission and the overhead charges are taken into consideration the economy is not so evident and may be completely neutralized. From a purely thermal standpoint, and as a means of conserving our natural resources, the producer-gas-electric plant is vastly superior to the ordinary steam-electric plant for power purposes, but the fuel item is only one of the many involved in the total cost. It is the *commercial efficiency* which enables the steam power plant, with its extravagant waste of fuel, to compete successfully with the gas producer, internal-combustion engine and hydro-electric plant.

A station which distributes power to a number of consumers more or less distant, is called a *Central Station*. When the distances are very great, electrical current of high tension is frequently employed, and is transformed and distributed at convenient points through *Substations*. A plant designed to furnish power or heat to a building or a group of buildings under one management is called an *Isolated Station*. For example, the power plant of an office building is usually called an isolated station.

When the exhaust steam from the engines is discharged at approximately atmospheric pressure the plant is said to be operating *non-*

condensing. When the exhaust steam is condensed, reducing the back pressure on the piston by the partial vacuum thus formed, the plant is said to operate *condensing.*

When the exhaust steam may be used for manufacturing, heating, or other useful purposes, as is frequently the case in various manufacturing establishments, and in large office buildings, it is usually more economical to run non-condensing, while power plants for electric lighting and power, pumping stations, air-compressor plants, and others, in which the load is fairly constant and the exhaust steam is not required for heating, are generally operated condensing.

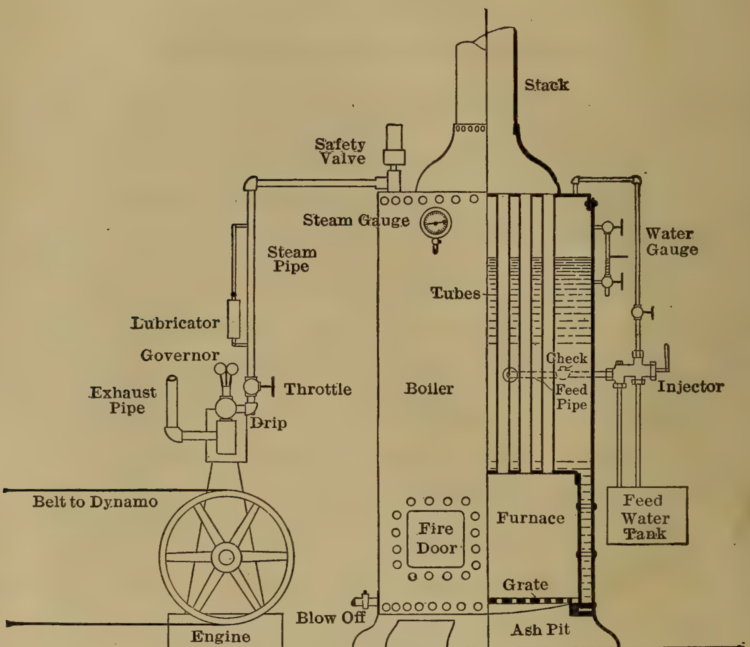


FIG. 1. Elementary Non-condensing Plant.

2. Elementary Non-condensing Plant. — Fig. 1 gives a diagrammatic outline of the essential elements of the simplest form of steam power plant. The equipment is complete in every respect and embodies all the accessories necessary for successful operation. Where a small amount of power is desired at intermittent periods, as in hoisting systems, threshing outfits and traction machinery, the arrangement is substantially as illustrated. The output in these cases seldom exceeds 50 horse power and the time of operation is usually short, so the cheapest of appliances are installed, simplicity and low first cost being more important than economy of fuel.

Such a plant has three essential elements: (1) The furnace, (2) the boiler, and (3) the engine. Fuel is fed into the *furnace*, where it is burned. A portion of the heat liberated from the fuel by combustion is absorbed by the water in the *boiler*, converting it into steam under pressure. The steam being admitted to the cylinder of the *engine* does work upon the piston and is then exhausted through a suitable pipe to the atmosphere. The process is a continuous one, fuel and water being fed into the furnace and the boiler in proportion to the power demanded.

In such an elementary plant, certain *accessories* are necessary for successful operation. The *grate* for supporting the fuel during combustion consists of a cast-iron grid or of a number of cast-iron bars spaced in such a manner as to permit the passage of air through the fuel from below. The solid waste products fall through or are "sliced" through the grate bars into the *ash pit*, from which they may be removed through the *ash door*. The latter acts also as a means of regulating the supply of air below the grate. Fuel is fed into the furnace through the *fire door*, and when occasion demands, air may be supplied above the bed of fuel by means of this door. The *combustion chamber* is the space between the bed of fuel and the boiler heating surface, its office being to afford a space for the oxidation of the combustible gases from the solid fuel before they are cooled below ignition temperature by the comparatively cool surfaces of the boiler. The *chimney* or *stack* discharges the products of combustion into the atmosphere and serves to create the draft necessary to draw the air through the bed of fuel. Various forced-draft appliances are sometimes used to assist or to entirely replace the chimney. The *heating surface* is that portion of the boiler area which comes into contact with the hot furnace gases, absorbs the heat and transmits it to the water. In the small plant illustrated in Fig. 1, the major portion of the heating surface is composed of a number of fire tubes below the water line, through which the heated gases pass. The *superheating surface* is that portion of the heating surface which is in contact with the heated gases of combustion on one side and steam on the other. The volume above the water level is called the *steam space*. Water is forced into the boilers either by a *feed pump* or an *injector*. In small plants of the type considered, steam pumps are seldom employed; the injector answers the purpose and is considerably cheaper. A *safety valve* connected to the steam space of the boiler automatically permits steam to escape to the atmosphere if an excessive pressure is reached. The water level is indicated by *try cocks* or by a *gauge glass*, the top of which is connected with the steam space and the bottom with the water space. *Try cocks* are small valves

placed in the water column or boiler shell, one at normal water level, one above it, and one below. By opening the valves from time to time the water level is approximately ascertained. They are ordinarily used in case of accident to the gauge glass. *Fusible plugs* are frequently inserted in the boiler shell at the lowest permissible water level. They are composed of an alloy having a low fusing point which melts when in contact with steam, thus giving warning by the blast of the escaping steam if the water level gets dangerously low. The *blow-off cock* is a valve fitted to the lowest part of the boiler to drain it of water or to discharge the sediment which deposits in the bottom. The steam outlet of a boiler is usually called the *steam nozzle.*

The essential accessories of the simple steam engine include: A *throttle valve* for controlling the supply of steam to the engine; the *governor,* which regulates the speed of the engine by governing the steam supply; the *lubricator,* attached to the steam pipe, which is usually of the "sight-feed" class and provides for lubrication of piston and valve. Lubrication of the various bearings is effected by *oil cups* suitably located. *Drips* are placed wherever a water pocket is apt to form in order that the condensation may be drained. The apparatus to be driven by the engine may be *direct connected* to the crank shaft or *belted* to the *flywheel* or *geared.*

In small plants of this type no attempt is made to utilize the exhaust steam except in instances where the stack is too short to create the necessary draft, in which case the exhaust may be discharged up the stack. If the draft is produced by convection of the heated gases in the chimney, the fuel is said to be burned under *natural draft;* if the natural draft is assisted by the exhaust steam, the fuel is said to be burned under *forced draft.* The power realized from a given weight of fuel is very low and seldom exceeds $2\frac{1}{2}$ per cent of the heat value of the fuel. The distribution of the various losses in a plant of, say, 40 horse power is approximately as follows:

	B.T.U.
Heat value of 1 pound of coal	14,500
Boiler and furnace losses, 50 per cent	7,250
Heat of the steam, 50 per cent.	7,250
Heat equivalent of one horse power hour	2,545
Heat used to develop one horse power hour (50 pounds steam per horse power hour, pressure 80 pounds gauge, feed water 62 degrees F.)	57,500
	Per cent.
Percentage of heat in the steam, realized as work, $\frac{2,545}{57,500}$	4.4
Percentage of heat value of the coal realized as work, $\frac{2,545}{57,500 \div 0.50}$	2.2

In Europe small non-condensing plants are developed to a high degree of efficiency. Through the use of highly superheated steam, specially designed engines and boilers, plants of this type as small as 40 horse power are operated with over-all efficiencies of from 10 to 12 per cent.

The power plant of the modern locomotive is very much like that illustrated in Fig. 1, the main difference lying in the type of boiler and engine. The entire exhaust from the engine is discharged up the stack through a suitable nozzle, since the extreme rate of combustion requires an intense draft. The engine is a highly efficient one compared with that in the illustration, and the performance of the boiler is more economical. In average locomotive practice about 6 per cent of the heat value of the fuel is converted into mechanical energy at the draw bar. In general, a non-condensing steam plant in which the heat of the exhaust is wasted is very uneconomical of fuel, even under the most favorable conditions, and seldom transforms as much as 7 per cent of the heat value of the fuel into mechanical energy.

3. Non-condensing Plant. Exhaust Steam Heating. — Fig. 2 gives a diagrammatic arrangement of a simple non-condensing plant differing from Fig. 1 in that the exhaust steam is used for heating purposes. This shows the essential elements and accessories, but omits a number of small valves, by-passes, drains, and the like for the sake of simplicity. The plant is assumed to be of sufficient size to warrant the installation of efficient appliances. Steam is led from the boiler to the engine by the *steam main*. The moisture is removed from the steam before it enters the cylinder by a *steam separator*. The moisture drained from the separator is either discharged to waste or returned to the boiler. The exhaust steam from the engine is discharged into the *exhaust main* where it mingles with the steam exhausted from the steam pumps. Since the exhaust from engines and pumps contains a large portion of the cylinder oil introduced into the live steam for lubricating purposes, it passes through an *oil separator* before entering the heating system. After leaving the oil separator the exhaust steam is diverted into two paths, part of it entering the *feed-water heater* where it condenses and gives up heat to the feed water, and the remainder flowing to the heating system. During warm weather the engine generally exhausts more steam than is necessary for heating purposes, in which case the surplus steam is automatically discharged to the *exhaust head* through the *back-pressure valve*. The back-pressure valve is, virtually, a large weighted check valve which remains closed when the pressure in the heating system is below a certain prescribed amount, but which opens automatically when the pressure is greater than this

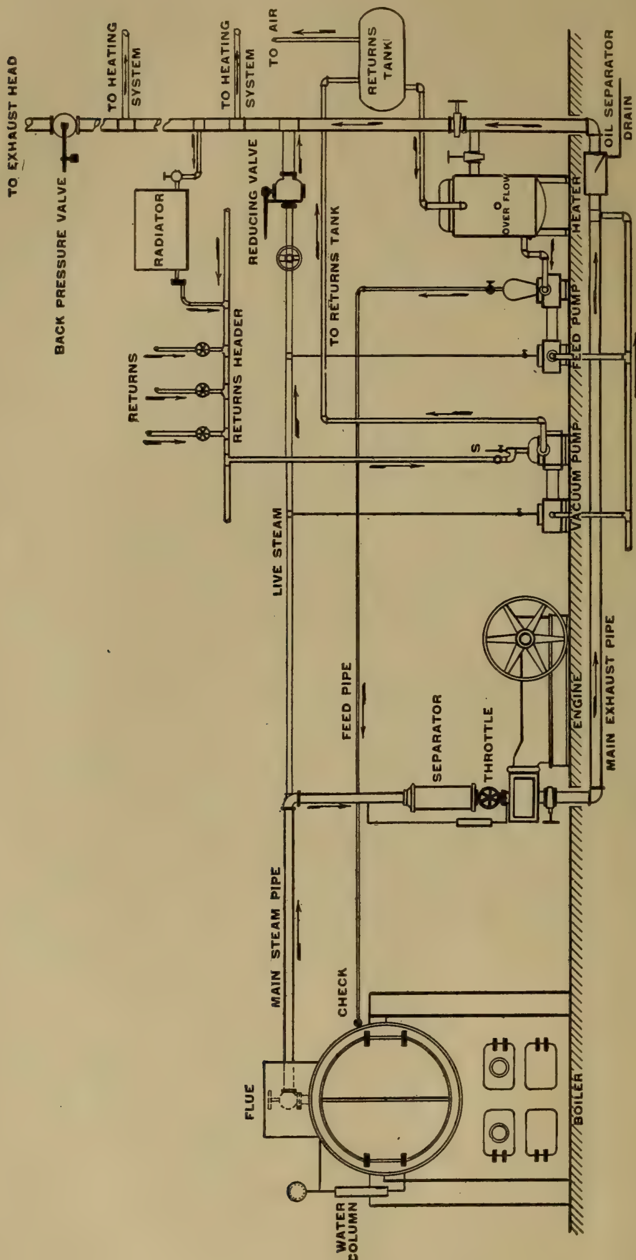


FIG. 2. Elementary Non-Condensing Plant with Heating System.

amount. During cold weather it often happens that the engine exhaust is insufficient to supply the heating system, the radiators condensing the steam more rapidly than it can be supplied. In this case live steam from the boiler is automatically fed into the main heating supply pipe through the *reducing valve.*

The condensed steam and the entrained air which is always present are automatically discharged from the radiators by a *thermostatic valve* into the *returns header.* The thermostatic valve is so constructed that when in contact with the comparatively cool water of condensation it remains open and when in contact with steam it closes. The *vacuum pump* or vapor pump exhausts the condensed steam and air from the returns header and discharges them to the *returns tank.* The small pipe S admits cold water to the vacuum pump and serves to condense the heated vapor, and at the same time supply the necessary *make-up water* to the system. The returns tank is open to the atmosphere so that the air discharged from the vacuum pump may escape. From the returns tank the condensed steam gravitates to the *feed-water heater* where its temperature is raised to practically that of the exhaust steam. The feed water gravitates to the *feed pump* and is forced into the boiler. There are several systems of exhaust steam heating in current practice which differ considerably in details, but, in a broad sense, are similar to the one just described. The more important of these will be described later on.

During the summer months when the heating system is shut down, the plant operates as a simple non-condensing station and practically all of the exhaust steam, amounting to perhaps 60 per cent of the heat value of the fuel, is wasted. The total coal consumption, therefore, is charged against the power developed. During the winter months, however, all, or nearly all, of the exhaust steam may be used for heating purposes and the power becomes a relatively small percentage of the total fuel energy utilized. The percentage of heat value of the fuel chargeable to power depends upon the size of the plant, the number and character of engines and boilers, and the conditions of operation. It ranges anywhere from 50 to 100 per cent for the summer months and may run as low as 6 per cent for the winter months. This is on the assumption, of course, that the engine is debited only with the difference between the coal necessary to produce the heat entering the cylinder and that utilized in the heating system.

4. Elementary Condensing Plant. — Under the most favorable conditions a non-condensing plant can never be expected to realize more than 7 per cent of the heat value of the fuel as power. In large non-condensing power stations the demand for exhaust steam is usually

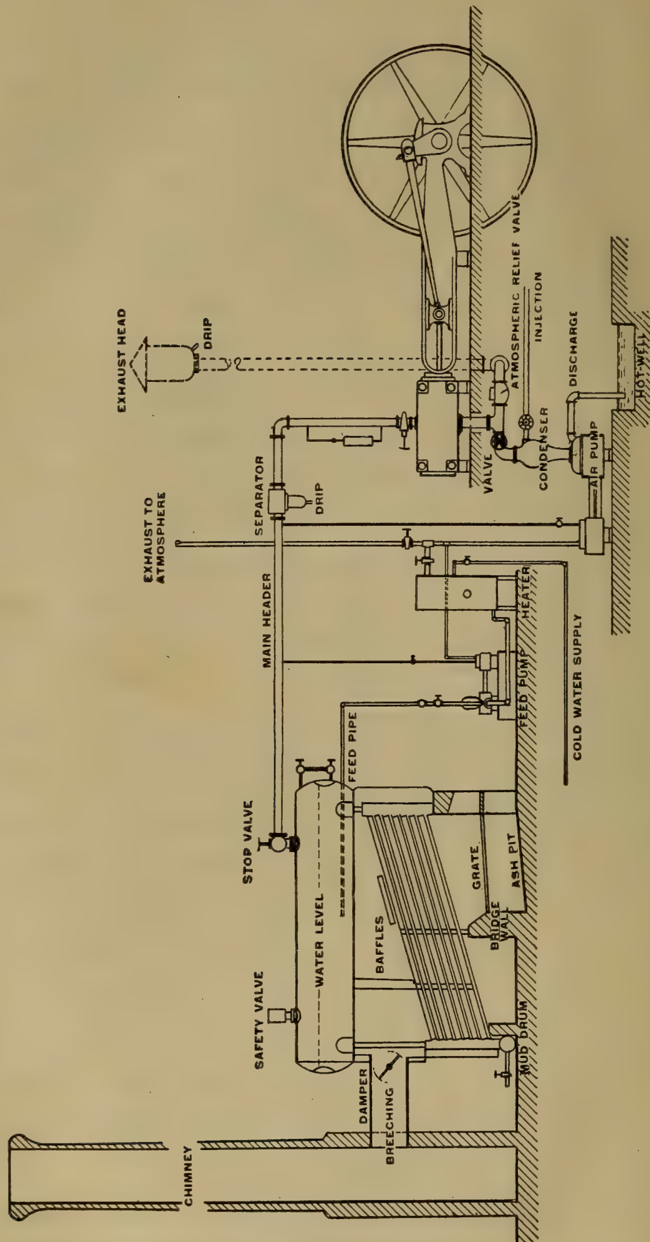


FIG. 3. Simple Condensing Plant.

limited to the heating of the feed water, and as only 12 or 15 per cent can be utilized in this manner, the greater portion of the heat in the exhaust is lost. Non-condensing engines require from 20 to 60 pounds of steam per hour for each horse power developed. On the other hand in condensing engines the steam consumption may be reduced to as low as 10 pounds per horse-power hour. The saving of fuel is at once apparent.

Fig. 3 gives a diagrammatic arrangement of a simple condensing plant in which the back pressure on the engine is reduced by condensing the exhaust steam. A different type of boiler from that in Fig. 1 or Fig. 2 has been selected, for the purpose of bringing out a few of the characteristic elements. The products of combustion instead of passing directly through *fire tubes* to the stack as in Fig. 1 are deflected back and forth across a number of *water tubes*, by the *bridge wall* and a series of *baffles*. After imparting the greater part of their heat to the heating surface the products of combustion escape to the chimney through the *breeching* or *flue*. The rate of flow is regulated by a *damper* placed in the breeching as indicated.

The steam generated in the boiler is led to the engine through the *main header*. The steam is exhausted into a *condenser* in which its latent heat is absorbed by *injection* or cooling water. The process condenses the steam and creates a partial vacuum. The condensed steam, injection water, and the air which is invariably present are withdrawn by an *air pump* and discharged to the *hot well*. In case the vacuum should fail, as by stoppage of the air pump, the exhaust steam is automatically discharged to the *exhaust head* by the *atmospheric relief valve*, and the engine will operate non-condensing. The *atmospheric relief valve* is a large check valve which is held closed by atmospheric pressure as long as there is a vacuum in the condenser. When the vacuum fails the pressure of the exhaust becomes greater than that of the atmosphere and the valve opens.

The feed water may be taken from the hot well or from any other source of supply and forced into the *heater*. In this particular case it is taken from a cold supply and upon entering the heater is heated by the exhaust steam from the *air* and *feed pumps*. From the heater it gravitates to the feed pump and is forced into the boiler. Various other combinations of heaters, pumps, and condensers are necessary in many cases, depending upon the conditions of operation. Feed pumps, air pumps, and in fact all small engines used in connection with a steam power plant are usually called *auxiliaries*.

A well-designed station similar to the one illustrated in Fig. 3 is capable of converting about 10 per cent of the heat value of the fuel

into mechanical energy. The various heat losses are approximately as follows:

BOILER LOSSES.	Per Cent.
Loss due to fuel falling through the grate	2
Loss due to incomplete combustion	2
Loss to heat carried away in chimney gases	23
Radiation and other losses	8
Total	35

	B.t.u.
Heat used by engines and auxiliaries (16 pounds of steam per i.h.p.-hour, pressure 150 pounds, feed water 210° F.) . . .	16,250
Engine and generator friction, 5 per cent	812
Leakage, radiation, etc., 2 per cent	325
Total	17,387
Heat equivalent of one electrical horse power	2,545

	Per Cent.
Percentage of the heat value of the steam converted into electrical energy	14.7
Percentage of heat value of fuel converted into electrical energy $\frac{2545 \times 0.65}{17,387}$	9.5

In large central stations equipped with turbo-generators using superheated steam, an over-all efficiency from switchboard to coal pile of 12 per cent is not unusual with a maximum of about 14 per cent.

5. Condensing Plant with Full Complement of Heat-saving Appliances. — When fuel is costly it frequently becomes necessary for the sake of economy to reduce the heat wastes as much as possible. The chimney gases, which in average practice are discharged at a temperature between 450 and 550 degrees F., represent a loss of 20 to 30 per cent of the total value of the fuel. If part of the heat could be reclaimed without impairing the draft the gain would be directly proportional to the reduction in temperature of the gases. Again, in some types of condensers all of the steam exhausted by the engine is condensed by the circulating water and discharged to waste. If provision could be made for utilizing part of the exhaust steam for feed-water heating, the efficiency of the plant could be correspondingly increased. In many cases the cost of installing such heat-saving devices would more than offset the gain effected, but occasions arise where they give marked economy.

Fig. 4 gives a diagrammatic arrangement of a condensing plant in which a number of heat-reclaiming devices are installed. The plant is assumed to consist of a number of engines, boilers, and auxiliaries. Coal is automatically transferred from the cars *to coal hoppers* placed above the boiler, by a system of buckets and conveyors. These hoppers store the coal in sufficient quantities to keep the boiler in continuous

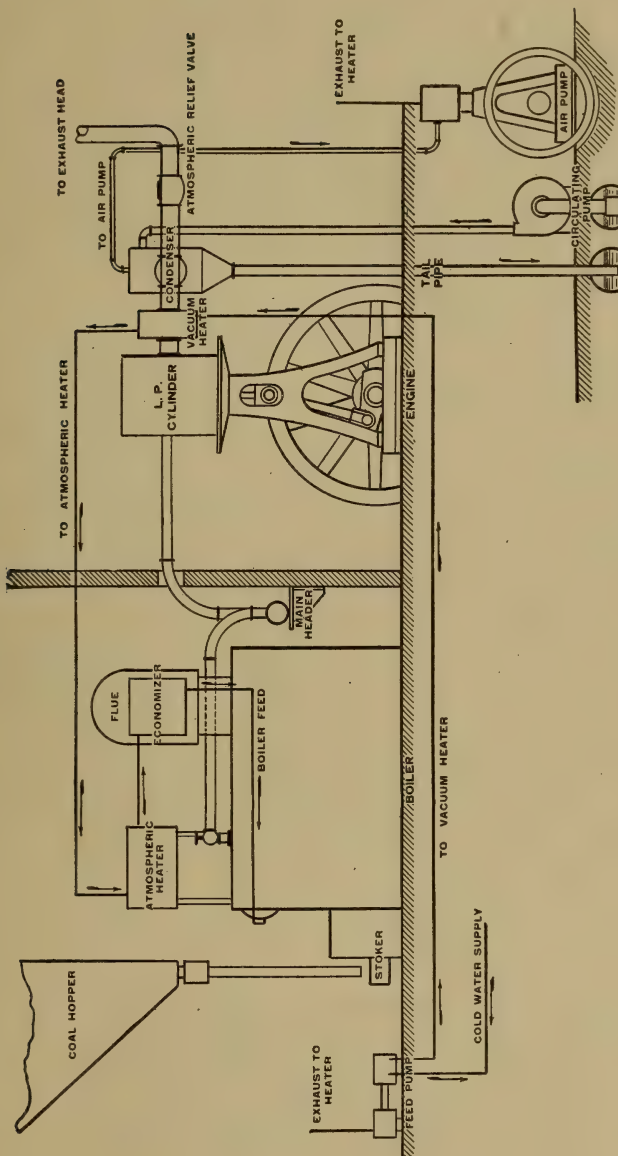


FIG. 4. Condensing Plant with Full Complement of Heat Saving Appliances.

operation for some time. From the hoppers the coal is fed intermittently to the *stoker* by means of a *down spout.* The stoker feeds the furnace in proportion to the power demanded and automatically rejects the ash and refuse to the *ash pit.* The ashes are removed from the ash pit, when occasion demands, and are transferred to the *ash hopper* by the same system of buckets and conveyor which handles the coal. The ash hopper is usually placed alongside the coal hoppers and is not unlike them in general appearance and construction.

The products of combustion are discharged to the stack through the flue or breeching. Within the flue is placed a feed-water heater called an *economizer,* the function of which is to absorb part of the heat from the gases on their way to the chimney. The heat reclaimed by the economizer varies widely with the conditions of operation and ranges between 5 and 20 per cent. Since the economizer acts as a resistance to the passage of the products of combustion it is sometimes necessary to increase the draft either by increasing the height of the chimney or, as is the usual practice, by using a forced-draft system.

Part of the heat of the exhaust steam is reclaimed by a *vacuum heater* which is placed in the exhaust line between engine and condenser. For example, if the feed water has a normal temperature of 60 degrees F. and the vacuum in the condenser is 26 inches, the vacuum heater will raise the temperature of the feed to, say, 120 degrees F., thereby effecting a gain in heat of approximately 6 per cent. If the feed supply is taken from the *hot well* the vacuum heater is without purpose, as the temperature of the hot well will not be far from 120 degrees F.

Referring to the diagram, the path of the steam is as follows: From the boiler it flows through the *boiler lead* to the *main header* or equalizing pipe. From the main header it flows through the *engine lead* to the high-pressure cylinder. The exhaust steam discharges from the low-pressure cylinder through the vacuum heater and into the condenser. Part of the exhaust steam is condensed in the vacuum heater and gives up its latent heat to the feed water. The remainder is condensed by the injection water which is forced into the condenser chamber by the *circulating pump.* The condensed steam and circulating water gravitate through the *tail pipe* to the hot well. The air which enters the condenser either as leakage or entrainment is withdrawn by the air pump. The steam exhausted by the feed pump, air pump, stoker engine, and other steam-driven auxiliaries is usually discharged into the *atmospheric heater,* which still further heats the feed water.

Referring to the feed water, the circuit is as follows: The pump draws in cold water at a temperature of, say, 60 degrees F., and forces it in turn through the vacuum heater, the atmospheric heater, and the

economizer into the boiler. The vacuum heater raises the temperature to 120 degrees F., the atmospheric heater increases it to 212 degrees F., and the economizer still further to about 300 degrees F. The heat reclaimed by this series of heaters is evidently the equivalent of that necessary to raise the feed water from 60 degrees F. to 300 degrees F., or approximately 24 per cent of the total steam supplied. In some plants the economizer only is installed; in others the economizer and atmospheric heater are deemed desirable; still others utilize all three. The distribution of the heat losses in a plant of this type using saturated steam and operating under favorable conditions is approximately as follows:

	Per Cent.	B.t.u.
Delivered to engine, 15 pounds steam per i.h.p.-hour; pressure 150 pounds, feed 60° F.	100	17,482
Delivered to feed pump	1.5	262
Delivered to circulating pump	1.5	262
Delivered to air pump	2	349
Delivered to small auxiliaries	1.5	262
Loss in leakage and drips	0.5	87
Engine and generator friction	5	874
Radiation and minor losses	1	175
Total		19,753

	Per Cent.	B.t.u.
Returned by vacuum heater	5.5	1,086
Returned by atmospheric heater	7.9	1,560
Returned by economizer	9.7	1,916
Total	23.1	4,562
Net heat delivered to engine in the form of steam to produce one electrical horse power, 19,753 − 4,562		15,191
Percentage converted to electrical power $\frac{2,545}{15,191}$	16.7	
Boiler efficiency	70	
Percentage of heat value of fuel necessary to produce one electrical horse power at switchboard $\frac{2545 \times 0.70}{15,191}$	11.7	

The preceding figures give the results of very good practice. So much depends upon the size and character of the prime movers, the nature of the fuel, and the conditions of operation that no definite figure can be given for the percentage of heat converted to power in a given type of station. Six per cent represents good average practice in a non-condensing plant and 10 per cent in a condensing plant using saturated steam. Pumping stations operating continuously under full load have realized as much as 15 per cent of the total heat value of the fuel, but such performances are practically unobtainable in connection

with steam-driven electrical power plants with the usual peak loads. Steam power plants as a class are very wasteful of fuel at the best.

One of the best recorded performances to date (March, 1912) of a steam-electric power plant is that of the Pacific Light and Power Company at Redondo, Cal. When operating under regular commercial conditions approximately 14 per cent of the available heat of the fuel (crude oil) is realized as power at the switchboard. This includes all standby losses. For a detailed description of the plant and the results of the acceptance tests, see Jour. of Elec. Gas and Power, Aug. 22, 1908.

In Europe a combined plant efficiency of 15 per cent is not uncommon. Even small semi-portable plants of 40 to 200 horse power are operated with over-all efficiencies as high as 14 per cent. In these small plants the engine, boiler, and auxiliaries are combined, permitting a high degree of superheat with minimum heat losses. A 40-horse-power plant tested by Professor Josse of the Royal Technical School, Germany, gave the following results: coal consumed per brake h.p.-hour, 1.23 pounds, corresponding to an over-all efficiency of 14.2 per cent. Steam consumption, 9.5 pounds per i.h.p.-hour. Boiler and superheated efficiency, 77.7 per cent. (See Zeit. des Ver. Deut. Ing., March 18 and 25, 1911, and Power, Sept. 27, 1910, p. 1714. See Fig. 226.)

CHAPTER II.

FUELS AND COMBUSTION.

6. General. — The subject of fuels and combustion has been so extensively treated by various authorities that a comprehensive discussion would be without purpose here, but in order to bring out more clearly the matter pertaining to the commercial design and operation of steam power plants a few of the essential elements will be briefly treated.

The fuels used for steam making are coal, coke, wood, peat, mineral oil, natural and artificial gases, refuse products such as straw, manure, sawdust, tan bark, bagasse, and occasionally corn and molasses.

In most cases that fuel is selected which develops the required power at the lowest cost, taking into consideration all of the circumstances that may affect its use. Occasionally the disposition of waste products is a factor in the choice, but such instances are uncommon. The boilers and furnaces are designed to suit the fuel selected.

7. Classification of Fuels. — Fuels may be divided into three classs as follows:

1. Solid fuels.
    a. Natural: straw, wood, peat, coal.
    b. Prepared: charcoal, coke, peat, and other briquettes.
2. Liquid fuels.
    a. Natural: crude oils.
    b. Prepared: distilled oils, alcohol, molasses.
3. Gaseous fuels.
    a. Natural: natural gas.
    b. Prepared: coal gas, water gas, producer gas, oil gas.

8. Solid Fuels. — Solid fuels are of vegetable origin and exist in a variety of forms between that of a comparatively recent cellulose growth and that of nearly pure carbon as anthracite coal. They owe their forms to the conditions under which they were created or to the geological changes which they have undergone. With each succeeding stage the percentage of carbon increases. The chemical changes are approximately as follows:

Substance.	Carbon.	Hydrogen.	Oxygen.
	Per Cent	Per Cent	Per Cent
Pure cellulose	44.44	6.17	49.39
Wood	52.65	5.25	42.10
Peat	59.57	5.96	34.47
Lignite	66.04	5.27	28.69
Brown coal	73.18	5.58	21.14
Bituminous coal	75.06	5.84	19.10
Semi-bituminous coal	89.29	5.05	6.66
Anthracite	91.58	3.96	4.46
Graphite	100.00	...	...

All natural solid fuels contain more or less earthy or inorganic matter which is not combustible and therefore remains as ash, while the organic matter is consumed. Sometimes the percentage of ash is so great as to render them valueless for steam-making purposes.

Origin and Formation of Fuel: Engng, Aug. 23, 1901; Am. Geol., Feb., 1899; Col. Guard, Sept. 10, 1897, Oct. 1, 1897, Jan. 14, 1898, Jan. 28, 1898, March 18, 1898, Sept. 14, 1900; Ec. Geol., Oct., 1905; Eng. U.S., April 1, 1903; Ir. and Coal Td. Review, Feb. 4, 1898, July 13, 1906.

9. Composition of Coal.— The uncombined carbon in coal is known as *fixed carbon*, while the hydrocarbons and other gaseous compounds which distill off on application of heat constitute the *volatile matter*. Refractory earths and moisture are found in varying quantities in different classes of coal and as they are incombustible tend to reduce the heat value of the fuel. That part of the fuel which is dry and free from ash is called the *combustible*, though the nitrogen and oxygen in the volatile matter are not actually combustible. The term "pure coal" has been suggested in this connection and is meeting with much favor. (Jour. W.S.E. 11–757.) The various elementary constituents of a fuel must be determined by a careful chemical analysis, but in most cases it is only necessary to know the heating value, the per cent of moisture and ash, and perhaps the per cent of sulphur. Tables 1 to 4 show the composition of a number of American coals and give a good idea of their chemical and physical characteristics.

10. Classification of Coals. — Coals and allied substances have been variously classified according to

1. Oxygen-hydrogen ratio, or Gruner's classification.
2. Fixed carbon and volatile combustible matter.
3. Fuel ratio, or the ratio of the fixed carbon to the volatile combustible matter.
4. Calorific value.
5. Fixed carbon.

6. Total carbon.
7. Hydrogen.
8. Carbon-hydrogen ratio, or the ratio of the total carbon to the hydrogen.

Gruner's classification is as follows:

(Eng. and Min. Jour., July 25, 1874.)

	Ratio $\frac{O}{H}$.		Ratio $\frac{O}{H}$.
Anthracite	1 to 0.75	Peat	6 to 5
Bituminous	4 to 1	Wood	7
Lignite	5	Cellulose	8

Kent's classification, according to the constituents of the combustible is as follows (Steam Boiler Practice):

	Per Cent of Dry Combustible.	
	Fixed Carbon.	Volatile Matter.
Anthracite	97 to 92.5	3 to 7.5
Semi-anthracite	92.5 to 87.5	7.5 to 12.5
Semi-bituminous	87.5 to 75	12.5 to 25
Bituminous — Eastern	75 to 60	25 to 40
Bituminous — Western	65 to 50	35 to 50
Lignite	Under 50	Over 50

Gruner's, Kent's, and the other schemes of classification outlined above, with the exception of the carbon-hydrogen ratio, are more or less unsatisfactory, since the groups are not as clearly defined as indicated and overlap to a considerable extent.

The U. S. Geological Survey proposes the following classification according to the carbon-hydrogen ratio which appears to apply satisfactorily to all grades of coal.

(Compiled from Report of Government Coal Testing Plant, Professional Paper No. 48, 1906.)

Group.	Class.	Example.	Carbon-hydrogen Ratio.
A	Graphite		
B	Anthracite	*Buck Mountain, Pa.	to 30
C	Anthracite	*Scranton, Pa.	30 to 26
D	Semi-anthracite	*Bernice Basin, Pa.	26 to 23
E	Semi-bituminous	Spadra Bed, Ark.	23 to 20
F	Bituminous	New River, W. Va.	20 to 17
G	...do	Connelsville Field, Pa.	17 to 14.4
H	...do	Marion County, Ill.	14.4 to 12.5
I	...do	Red Lodge, Mont.	12.5 to 11.2
J	Lignite	Gallup Field, N. M.	11.2 to 9.3
K	Peat		9.3 to
L	Wood		7.2

* Not included in Government's Report.

For a classification of various coals according to the calorific value, fixed carbon, total carbon and hydrogen content, consult government report.

Classification of Coals: Prac. Engr. U. S., Jan., 1910; Mines and Minerals, Feb., 1911; Am. Inst. Min. Engrs., May, 1906, Sept., 1905; Eng. Mag., Jan., 1912.

11. Anthracites. — These are the most perfect coals and consist almost entirely of carbon; they contain very little hydrocarbon and burn with little or no smoke, are slow to ignite, burn slowly, and break into small pieces when rapidly heated. They require a very large grate of about twice the surface necessary for bituminous coal. Large sizes may be burned in almost any kind of a furnace and with moderate draft.

TABLE 1.

COMPOSITION OF TYPICAL AMERICAN ANTHRACITE COALS.

	Treverton, Pa.	Wilkesbarre, Pa.	Drifton, Pa.	Lehigh, Pa.	Scranton, Pa., Culm, Air Dried.	Lykens Valley.
Proximate analysis:	*	*	†	†	‡	†
Water	0.84	3.45	1.37	1.97	2.08	1.50
Volatile matter	6.67	2.75	3.59	4.35	7.27	7.84
Fixed carbon	85.66	87.90	89.11	86.49	74.32	81.07
Ash	6.83	5.90	5.93	7.19	16.33	9.59
	100.00	100.00	100.00	100.00	100.00	100.00
Ultimate analysis:						
Carbon	90.66	88.86	87.70	85.66	75.21	83.20
Hydrogen	1.73	2.04	2.56	2.78	2.81	3.29
Nitrogen	....	0.90	1.03	0.77	0.80	0.95
Oxygen	0.78	1.95	2.26	2.87	4.08	2.45
Sulphur	....	0.35	0.56	0.64	0.77	0.50
Ash	6.83	5.90	5.89	7.28	16.33	9.61
	100.00	100.00	100.00	100.00	100.00	100.00
Calorific value:						
Calorimeter	13,980	13,950	....	....	12,472	....
Dulong's formula	14,194	14,103	14,217	14,038	12,426	14,003
Classification:						
Carbon-hydrogen ratio	52.5	42.5	34.4	30.9	26.7	25.
Fuel ratio	12.9	32.0	29.9	11.0	10.2	10.4

* Authority not stated. † H. J. Williams. ‡ U. S. Geological Survey.

For smaller sizes a thinner bed has to be carried unless a strong draft is used. There is difficulty in keeping a thin bed free from air-holes. When possible the coal should be at least six inches deep on the grate. On account of the large percentage of ash in the smaller

sizes, the fire requires frequent cleaning. Anthracites do not require "slicing" and should be disturbed only when cleaning is necessary. Nearly all anthracites, with some unimportant exceptions, come from three small fields in eastern Pennsylvania. On account of the limited supply and the great demand for domestic purposes, sizes over "pea coal" are prohibitive in price for steam power plant use. Table 1 gives the composition and classification of a number of typical American anthracite coals, and Table 2, one of the standard divisions of mesh according to which they are classed and marketed. Specific gravity, 1.4 to 1.6.

Burning No. 3 Buckwheat: Power, Dec. 27, 1910; Mar. 21, 1911. *Burning Anthracite Culm of Poor Quality:* Trans. A.S.M.E., 7–390. *Anthracite Culm Briquets,* Am. Inst. Min. Engrs., Bulletin, Sept., 1911. *Calorific Value of Anthracite:* Mines and Minerals, Sept., 1911. *Preparation of Anthracite:* Am. Inst. Min. Engrs., Bulletin, Oct., 1911.

TABLE 2.

STANDARD SIZES OF MESH: ANTHRACITE COAL.

(Lehigh Coal and Navigation Company, Philadelphia, Pa.)

Steamboat	Over	$4\frac{1}{2}$	Round.
Broken	Through	$4\frac{1}{2}$	Round.
	Over	$3\frac{1}{4}$	Round.
Egg	Through	$3\frac{1}{4}$	Round.
	Over	$2\frac{5}{16}$	Round.
Stove	Through	$2\frac{5}{16}$	Round.
	Over	$1\frac{10}{16}$	Round.
Nut	Through	$1\frac{10}{16}$	Round.
	Over	$\frac{14}{16}$	Round or $\frac{12}{16}$ square.
Pea	Through	$\frac{14}{16}$	Round or $\frac{12}{16}$ square.
	Over	$\frac{9}{16}$	Round.
Buckwheat No. 1	Through	$\frac{9}{16}$	Round.
	Over	$\frac{5}{16}$	Round.
Buckwheat No. 2	Through	$\frac{5}{16}$	Round.
	Over	$\frac{3}{16}$	Round.
Buckwheat No. 3	Through	$\frac{3}{16}$	Round.
	Over	$\frac{3}{32}$	

12. Semi-anthracites. — These coals kindle more readily and burn more rapidly than the anthracites. They require little attention, burn freely with a short flame and yield great heat with little clinker and ash. They are apt to split up on burning and waste somewhat in falling through the grate. They swell considerably, but do not cake. They have less density, hardness and metallic luster than anthracite, and can generally be distinguished by their tendency to soil the hands, while pure anthracite will not. Semi-anthracites are not of great importance in the steam power plant field on account of the limited supply and high cost. They are found in a few small areas in the western part of the anthracite field. Specific gravity, 1.3 to 1.4.

13. Semi-bituminous. — These coals are similar in appearance to semi-anthracite, but they are somewhat softer and contain more volatile matter. They have a very high heating value, have a low moisture, ash and sulphur content, are readily burned without producing objectionable smoke and rank among the best steaming coals in the world. The supply is limited and on account of high cost, except in the immediate vicinity of the mines, they are not generally used for power purposes. Table 3 gives the composition and classification of a number of typical American semi-anthracite and semi-bituminous coals.

TABLE 3.

COMPOSITION OF TYPICAL AMERICAN SEMI-ANTHRACITE AND SEMI-BITUMINOUS COALS.

	Bernice Basin, Pa.	Coalhill, Ark.	Pocahontas, W. Va.	Kanawha Series, W. Va.	New River, W. Va.	Clearfield, Pa.
	*	*	†	‡	†	§
Proximate analysis:						
Water	1.57	2.36	4.07	1.42	1.53	0.44
Volatile matter	9.40	12.68	16.34	20.72	21.54	18.76
Fixed carbon	83.69	72.88	68.47	70.05	71.88	73.15
Ash	5.34	12.08	11.12	7.81	5.05	7.65
	100.00	100.00	100.00	100.00	100.00	100.00
Ultimate analysis:						
Carbon	85.46	76.44	76.51	81.95	82.87	80.32
Hydrogen	3.72	3.82	4.27	4.30	4.76	4.88
Nitrogen	1.12	1.37	1.00	1.29	1.68	1.46
Oxygen	3.45	4.30	6.59	3.68	4.99	4.69
Sulphur	0.91	1.99	0.51	0.97	0.65	1.00
Ash	5.34	12.08	11.12	7.81	5.05	7.65
	100.00	100.00	100.00	100.00	100.00	100.00
Calorific value:						
Calorimeter	....	13,259	13,509	14,686	14,807	....
Dulong's formula	14,552	13,273	13,329	14,363	14,691	14,432
Classification:						
Carbon-hydrogen ratio‖	23.0	20.7	19.6	19.0	17.8	16.5
Fuel ratio	8.5	5.7	4.2	3.4	3.3	3.9

* Authority not stated. † U. S. Geological Survey. ‡ W. Va. Geological Survey. § H. J. Williams. ‖ Based on air-dried sample.

14. Bituminous. — These coals are the most widely distributed and the most extensively used fuel in steam power plant engineering. They contain a large and varying amount of volatile matter and burn freely with the production of considerable smoke unless carefully fired. Their physical properties vary widely and they are commonly classified as

1. Dry, or free-burning bituminous.
2. Bituminous caking.
3. Long-flaming bituminous.

TABLE 4.

COMPOSITION OF TYPICAL AMERICAN BITUMINOUS COALS.*

	Freeport, W. Va.	Hocking Valley, Ohio.	Pere Marquette, Mich.	Brazil, Ind.	Eastern Field, Kentucky.	Horsecreek, Ala.	Fleming, Kansas.	Laddadale, Iowa.	Rich Hill, Mo.	Coffeen, Ill.	Cambria, Wyo.	Chariton, Iowa.
Proximate analysis:		†	†	†								
Water	1.48	6.65	12.15	8.98	3.10	2.34	4.99	8.24	8.33	14.43	9.44	15.39
Volatile matter	28.58	34.14	31.14	34.49	36.12	31.84	32.68	30.74	33.58	29.48	35.02	30.49
Fixed carbon	61.55	49.54	53.95	50.30	56.39	53.28	49.36	45.02	38.73	42.81	34.82	41.49
Ash	8.39	9.67	2.76	6.33	4.39	12.54	12.97	16.00	19.36	13.28	20.72	12.63
	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Ultimate analysis:												
Carbon	77.82	70.86	79.44	70.50	77.37	71.58	67.34	59.82	57.00	54.59	51.46	55.81
Hydrogen	4.89	4.70	5.29	4.76	5.43	5.01	4.98	4.81	4.97	5.49	5.00	5.74
Nitrogen	1.48	1.53	1.56	1.36	1.83	1.65	1.08	0.94	0.94	1.11	0.74	1.14
Oxygen	6.52	11.45	9.84	15.66	9.76	8.50	9.35	13.40	12.48	21.52	18.17	21.49
Sulphur	0.90	1.79	1.11	1.39	1.22	0.72	4.28	5.03	5.25	4.01	3.91	3.19
Ash	8.39	9.67	2.76	6.33	4.39	12.54	12.97	16.00	19.36	13.28	20.72	12.63
	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Calorific value:												
Calorimeter	14,069	13,053	....	12,417	14,148	12,850	12,242	11,027	10,586	10,064	9650	10,242
Dulong's formula	13,927	12,444	14,160	12,084	13,955	12,927	12,217	10,881	10,656	9,866	9362	10,173
Classification:												
Carbon-hydrogen ratio‡	16.10	15.10	15.00	14.80	14.60	14.50	13.90	13.40	12.90	12.30	12.20	11.20
Fuel ratio	2.16	1.46	1.73	1.46	1.56	1.83	1.51	1.47	1.15	1.45	0.99	1.36

* Compiled from Report of Government Coal Testing Plant, U. S. Geological Survey.

† Not included in government report. ‡ Based on air-dried sample.

1. Dry bituminous coals are the best of the bituminous variety for steaming purposes. They are hard and dense, black in color, but somewhat brittle and splintery. They ignite readily, burn freely with a short clean bluish flame and without caking. Specific gravity, 1.25 to 1.40.

2. Bituminous caking coals swell up, become pasty and fuse together in burning. They contain less fixed carbon and more volatile matter than the free-burning grades. Caking coals are rich in hydrocarbon and are particularly adapted to gas making. The flame is of a yellowish color. Specific gravity, about 1.25.

3. Long-flaming bituminous coals are similar in many respects to the caking coals but contain a larger percentage of volatile matter. They burn freely with a long yellowish flame. They may be either caking, non-caking or splintery. They are very valuable as a gas coal, and are little used for steaming purposes. Specific gravity, about 1.2.

Table 4 gives the composition and classification of a number of typical American bituminous coals.

For sizes of bituminous coal see paragraph 31.

COAL FIELDS OF THE UNITED STATES.

Alabama: Mines and Minerals, May, 1901.

Alaska: Mining World, Aug. 28, 1909; Eng. and Min. Jour., Aug. 6, 1910.

Arizona: Trans. Am. Inst. Min. Engrs., Feb. and May, 1902.

California: Eng. and Min. Jour., July 4, 1896; Mining World, Feb. 17, 1906.

Colorado: Mines and Minerals, May, 1905, May, 1910; Min. Rept., Jan. 19, 1905.

Idaho: Eng. and Min. Jour., March 5, 1910; Mines and Mining, Jan., 1903.

Illinois: Min. Mag., March, 1905; Eng. and Min. Jour., Jan. 13, 1906.

Iowa: Mines and Mining, Sept., 1910; Eng. and Min. Jour., May 10, 1902.

Kentucky: Eng. and Min. Jour., Apr. 27, 1907; Jan. 18, 1908; June 27, 1908; Aug. 14, 1909.

Maryland: U. S. Geol. Survey, Annual Report, 1902, part 3.

Michigan: Eng. and Mining Jour., June 30, 1900; Min. World, Feb. 9, 1907.

Mississippi: Eng. and Min. Jour., Jan. 16, 1909.

Missouri: Am. Inst. of Min. Engrs., Jan., 1905.

Montana: Min. Mag., March, 1905; Min. World, Nov. 24, 1906.

Nebraska: Eng. and Min. Jour., Vol. 73, p. 481.

Nevada: Eng. and Min. Jour., Dec. 31, 1910.

New Mexico: Eng. and Min. Jour., May 21, 1910; June 20, 1908; March 7, 1908.

North Carolina: Eng. and Min. Jour., June 11, 1910; Aug. 25, 1906.

North Dakota: Eng. and Min. Jour., Jan. 15, 1910; April 10, 1909.

Ohio: Min. Mag., Mar., 1905; U. S. Geol. Survey, part 3, 1902.

Oklahoma (Ind. Terr.): Min. Rept., May 17, 1906; Mining World, Dec. 12, 1908.

Oregon: Eng. and Min. Jour., Aug. 17, 1907; Feb. 15, 1902.

Pennsylvania: Eng. and Min. Jour., Aug. 24, 1901; Pro. Eng. S. W. Penn., Jan., 1907.

Rhode Island: Ry. Age Gazette, July 8, 1910.

Tennessee: Eng. and Min. Jour., April 1, 1911; Mines and Mining, Sept., 1910.
Texas: Mines and Minerals, Oct., 1905.
Utah: Mines and Mining, Sept., 1906; June, 1909.
Virginia: Mines and Minerals, March, 1906; Eng. News, Oct. 20, 1904.
Washington: Eng. and Min. Jour., Aug. 19, 1911; U. S. Geol. Survey, part 3, 1902.
West Virginia: Eng. and Min. Jour., May 12, 1904.
Wisconsin: Trans. Am. Inst. of Mech. Engrs., Vol. 8, p. 478.
Wyoming: Mining World, May 6, 1905; Coal Age, Apr. 13, 1912.

GENERAL.

Coal Mines of the United States: Peabody Atlas, A. Bement, Chicago, Ill., Min. World, May 6, 1905; Eng. and Min. Jour., Jan. 8, 1910.
Coal Resources of the Pacific: Eng. Mag., May, 1902.
Rocky Mountain Coal Fields: Min. Rept., Jan. 5, 1905; Jour. Asso. Eng. Soc., Dec., 1902.
Coal Fields, U. S. Northwest: Rev. of Rev., Feb., 1903.
Coal Fields, U. S. Southwest: Eng. and Min. Jour., Oct. 17, 1903.
Report of Coal Testing Plant: U. S. Geological Survey, Washington, D. C. (1906).
Index of Mining Engineering Literature: W. R. Crane, John Wiley & Sons.

15. Lignite, or brown coal, is a substance of more recent geological formation than coal and represents a stage in development intermediate between coal and peat. Its specific gravity is low, 1.2, and when freshly mined contains as high as 50 per cent of moisture. It is non-caking, and on exposure to air, slackens or crumbles. The lumps check and fall into small irregular pieces with a tendency to separate into extremely thin plates. It deteriorates greatly during storage or long transportation. Lignite, as mined, is a low-grade fuel with a calorific value of about one-half that of good coal. When properly prepared and compressed into briquettes lignite becomes an excellent fuel, resists weathering satisfactorily, permits handling and transportation without excessive deterioration and is practically smokeless. The superiority of briquettes over raw lignite is shown by the following table:

IMPROVEMENT OF HEAT VALUE BY BRIQUETTING.*

Source	Moisture			Heat Value per Pound.		
	In Raw Lignite.	In Briquettes.	Removed.	Raw Lignite.	Briquettes.	Increase.
	Per Cent.	Per Cent.	Per Cent.	B.t.u.	B.t.u.	Per Cent.
Texas	33.0	9.0	24.0	6840	9336	36.5
North Dakota	40.0	12.0	28.0	6241	9354	50.0
North Dakota	42.0	10.0	32.0	6079	9355	54.0
California	40.0	10.0	30.0	6080	9264	52.4

* Bulletin No. 14, U. S. Bureau of Mines, p. 48.

The most extensive lignite deposits are situated long distances from fields of high-grade coal, and although the use of lignite is at present limited to these regions it is fast becoming a general competitor of coal.

North Dakota Lignite as a Fuel for Power Plant Boilers: Bul. No. 2, 1910, U. S. Bureau of Mines. *Briquetting Tests of Lignite:* Bul. No. 14, 1911, U. S. Bureau of Mines. General data pertaining to lignite fuels, Engr. U. S., Jan., 1910.

TABLE 5.

COMPOSITION AND CLASSIFICATION OF TYPICAL AMERICAN LIGNITES.*

(Run of Mine.)

	Red Lodge, Montana. (Black.)	Gallup, New Mexico. (Black.)	Texas. (Brown.)	Colorado. (Black.)	North Dakota. (Brown.)	Wyoming. (Black.)
Proximate analysis:						
Water	11.05	12.29	33.71	18.68	36.78	22.63
Volatile matter	35.90	34.58	29.25	34.88	28.16	35.68
Fixed carbon	42.08	46.14	29.76	40.45	29.97	37.19
Ash	10.97	6.99	7.28	5.99	5.09	4.50
	100.00	100.00	100.00	100.00	100.00	100.00
Ultimate analysis:						
Hydrogen	5.37	5.82	6.79	6.07	6.93	6.39
Carbon	59.08	63.31	45.52	57.46	41.87	54.91
Nitrogen	1.33	1.03	0.79	1.15	0.69	1.02
Oxygen	21.52	22.22	42.09	28.78	44.94	32.59
Sulphur	1.73	0.63	0.53	0.55	0.48	0.59
Ash	10.97	6.99	7.28	5.99	5.09	4.50
	100.00	100.00	100.00	100.00	100.00	100.00
Calorific value:						
Calorimeter	10,539	11,252	7348	10,143	7002	9734
Dulong's formula	10,355	11,153	7177	9,948	6944	9478
Classification:						
Carbon-hydrogen ratio†	11.50	11.20	10.90	9.80	9.60	9.40
Fuel ratio	1.17	1.09	1.02	1.16	1.06	1.05

* Compiled from Government Report, U. S. Geological Survey.

† Based on air-dried analysis.

16. Peat, or Turf, is formed by the slow carbonization under water of a variety of accumulated vegetable materials. It is unsuitable for fuel until dried. Peat, as ordinarily cut and dried, is too bulky for commercial competition with coal, and is used only where coal is prohibitive in price. When properly prepared and compressed into briquettes peat is an excellent fuel. In Russia, Germany, and Holland peat briquettes have passed the experimental stage and several millions of pounds are manufactured annually. Peat is used but little in this country at present, though the deposits are extensive and widely distributed, but its possibilities are beginning to attract the attention of

engineers. The proportion in which the various primary constituents exist in dried peat is approximately as follows:

	Per Cent.
Fixed carbon	35
Volatile matter	60
Ash	5

Peat: Prac. Engr. U. S., Jan., 1910, p. 21; Bul. No. 16, U. S. Bureau of Mines, 1911; Power, Sept. 6, 1910; Eng. and Min. Jour., Nov. 22, 1902; Feb. 7, 1903, Jour. Am. Peat Soc., July, 1911; Elec. Rev., Mar. 22, 1912; Min. and Eng. Wld., Nov. 28, 1911.

17. Wood, Straw, Sawdust, Bagasse, Tanbark. — In certain localities cordwood is still used as a fuel, but the steadily increasing values of even the poorest qualities are rapidly prohibiting its use for steam-

TABLE 6.

PHYSICAL AND CHEMICAL PROPERTIES OF WOODS, STRAW AND TANBARK.

(Prac. Engr. U. S., Jan., 1910.)

	Weight per Cubic Foot. Pounds.	Weight per Cord. Pounds.	Equivalent Weight of Coal. 13,500 B.T.U.	Carbon. Per Cent.	Hydrogen. Per Cent.	Oxygen. Per Cent.	Nitrogen. Per Cent.	Ash. Per Cent.	Calorific value, B.T.U. per Pound.	Authority.
Ash	46	3520	1420	......	......	......	......	......	5450	Hutton
Beech	43	3250	1300	49.36	6.01	42.69	0.91	1.06	5400	Sharpless
Birch	45	2880	1190	50.20	6.20	41.62	1.15	0.81	5580	Hutton
Cherry	42	3140	1260	......	......	......	......	......	5420	"
Chestnut	41	2350	940	......	......	......	......	......	5400	Sharpless
Elm	35	2350	940	......	......	......	......	......	5400	"
Hemlock	25	1220	580	......	......	......	......	......	6410	Hutton
Hickory	53	4500	1800	......	......	......	......	......	5400	Sharpless
Maple, Hard	49	3310	1340	......	......	......	......	......	5460	Hutton
Oak, Live	59	3850	1560	......	......	......	......	......	5460	"
" White	52	3850	1540	49.64	5.92	41.16	1.29	1.97	5400	Rankine
" Red	45	3310	1340	......	......	......	......	......	5460	Hutton
Pine, White	25	1920	970	......	......	......	......	......	6830	"
" Yellow	36	2130	1050	......	......	......	......	......	6660	"
Poplar	36	2130	1050	49.37	6.21	41.60	0.96	1.86	6660	"
Spruce	25	1920	970	......	......	......	......	......	6830	"
Walnut	35	3310	1340	......	......	......	......	......	5460	"
Willow	25	1920	970	49.96	5.96	39.56	0.96	3.37	6830	Rankine
Average	....	.....	......	49.70	6.06	41.30	1.05	1.80		
Straw:	6 to 8*		Water							
Wheat		.....	16.00	35.86	5.01	37.68	0.45	5.00	......	Clark
Barley		.....	15.50	36.27	5.07	38.26	0.40	4.50	......	"
Average			15.75	36.06	5.04	37.97	0.42	4.75	5155	
Tanbark Dry	....	.....	......	51.80	6.04	40.74	......	1.42	6100	Myers

* Compressed.

generating purposes. Sawdust, shavings, tanbark and other waste products of wood are burned under boilers in situations where such disposition nets the best financial returns. Recent progress, however, in industrial chemistry shows that ethyl and wood alcohols and other valuable by-products can be cheaply made from sawdust, shavings, slashings and similar waste material, and it is not unlikely that their use for steaming purposes will be unheard of in a comparatively few years. Table 6 gives the physical and chemical characteristics of a number of woods.

Wood as Fuel: Prac. Engr. U. S., Jan., 1910, p. 805; Power & Engr., June 30, 1908, p. 1015; Power, Dec., 1908, p. 772.

Burning Sawdust: Prac. Engr. U. S., Jan., 1910, p. 48; Power & Engr., April 7, 1908, p. 536; Oct. 13, 1908, p. 613; Jour. of Elec., Oct., 1905.

TABLE 7.

HEAT VALUES OF BAGASSE AND VARIATION WITH DEGREE OF EXTRACTION.

Per Cent Extraction on Weight of Cane.	Per Cent Moisture in Bagasse.	Fiber.		Sugar.		Molasses.		Total Heat developed. B.T.U.	Heat required to evaporate the Water present. B.T.U.	Heat available. B.T.U.	Lbs. Bagasse required to equal 1 lb. Coal of 14,000 B.T.U. Calorific Power.	Coal Equivalent per Ton of Cane. Pounds.	Temperature of Fire. Fahr.
		Per Cent in Bagasse.	Fuel Value. B.T.U.	Per Cent in Bagasse.	Fuel Value. B.T.U.	Per Cent in Bagasse.	Fuel Value, B.T.U.						
90	0.00	100.00	8325	....	....	....	....	8325	....	8325	1.68	119	2465°
85	28.33	66.67	5552	3.33	240	1.67	116	5900	339	5561	2.52	119	2236
80	42.50	50.00	4160	5.00	361	2.50	174	4697	509	4188	3.34	120	2023
75	51.00	40.00	3330	6.00	433	3.00	209	3972	611	3361	4.17	120	1862
70	56.67	33.33	2775	6.67	482	3.33	232	3489	679	2810	4.98	120	1732
65	60.71	28.57	2378	7.15	516	3.57	248	3142	727	2415	5.80	121	1612
60	63.75	25.00	2081	7.50	541	3.75	261	2883	764	2119	6.61	121	1513
55	66.12	22.22	1850	7.78	562	3.88	270	2682	792	1890	7.40	121	1427
50	68.00	20.00	1665	8.00	578	4.00	278	2521	815	1706	8.21	122	1350
45	69.55	18.18	1513	8.18	591	4.09	284	2388	833	1555	9.00	122	1284
40	70.83	16.67	1388	8.33	601	4.17	290	2279	849	1430	9.79	123	1222
25	73.67	13.33	1110	8.67	626	4.33	301	2037	883	1154	12.13	124	1077
15	75.00	11.77	980	8.82	637	4.41	307	1924	899	1025	13.66	124	1002
0	76.50	10.00	832	9.00	650	4.50	313	1795	916	879	15.93	126	906

Bagasse, or megass, is refuse sugar cane and is used as a fuel on the sugar plantations. Its heat value depends upon the proportions of fiber, molasses, sugar and water left after the extraction. The heat furnished by the different constituents is about as follows: Fiber, 8325 B.t.u. per pound; sugar, 7223 B.t.u. per pound; and molasses, 6956 B.t.u. per pound. Table 7 gives the heat value of bagasse and variation with the degree of extraction. A typical furnace for burning bagasse is shown in Fig. 5.

Bagasse as Fuel: Prac. Engr. U. S., Jan., 1910; Engng., Feb. 18, 1910.

Bagasse Drying: E. W. Kerr, Louisiana Bul. No. 128, June, 1911.

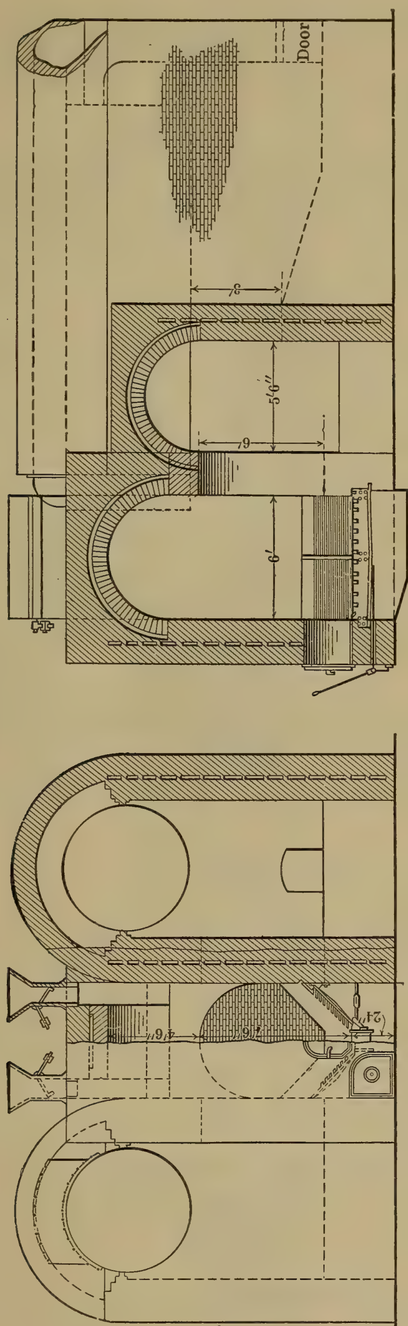


Fig. 5. Sectional Elevations of the Myers Furnace for Burning Bagasse.

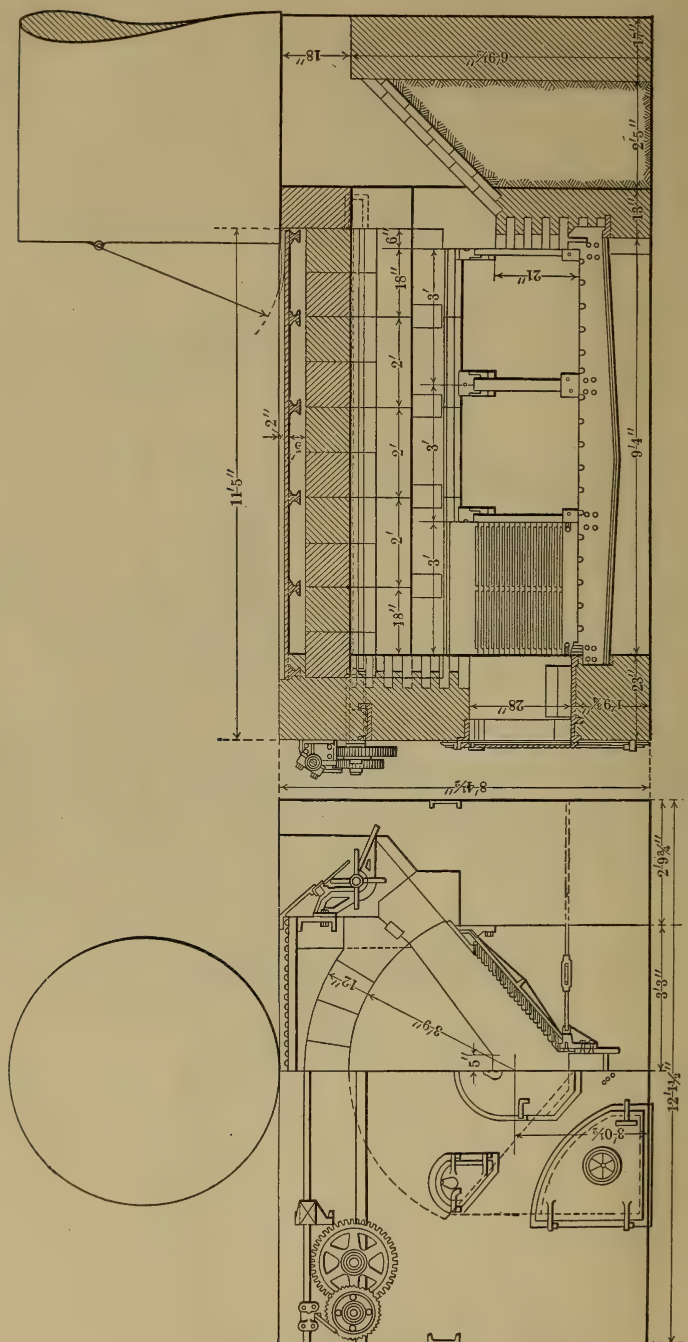


FIG. 6. End and Side Sectional Elevation of the Myers Tanbark Furnace.

Tanbark is usually quite moist; the amount of moisture varies with the leaching process used and averages around 65 per cent. In this condition it has a heat value of about 4300 B.t.u. per pound. If perfectly dry its heating power is approximately 6100 B.t.u. per pound. As in the case of all moist fuels, tanbark must be surrounded by heated surfaces of sufficient extent to insure drying out the fresh fuel as thrown on the fire. A successful furnace for burning tanbark is shown in Fig. 6.

Tanbark as a Boiler Fuel: Jour. A.S.M.E., Feb., 1910, p. 181; Jour. A.S.M.E., Oct., 1909, p. 951; Prac. Engr. U. S., Jan., 1910.

18. Combustion. — By combustion is meant the chemical union of the combustible material of a fuel and the oxygen of the air. Theoretically the process is a simple one, as it is only necessary to bring each particle of fuel previously heated to the kindling temperature in contact with the correct amount of oxygen and the combustion will be complete, the fuel oxidizing to the highest possible degree. In practice, however, the size and character of fuel, type of furnace, draft, impurities in the fuel, and the mechanical difficulties affect combustion to such an extent as to render oxidation more or less incomplete.

When heat is applied to coal, combustion takes place in three separate and distinct stages:

1. Absorption of heat. A fresh charge of fuel when thrown on a fire must first be brought to the kindling point in order that chemical action may take place. The temperatures necessary to cause this union of oxygen and fuel are approximately as follows:

	Degrees F.		Degrees F.
Lignite dust	300	Cokes	800
Sulphur	470	Anthracite lump	750
Dried peat	435	Carbon monoxide	1211
Anthracite dust	570	Hydrogen	1100
Lump coal	600		

(Stromeyer, Marine Boiler Management and Construction, p. 93.)

2. Vaporization of the hydrocarbon portion of the fuel and its combustion, the hydrocarbons consisting principally of olefiant gas, C_2H_4, marsh gas, CH_4, tar, pitch, naphtha and the like. As these gases are driven off they become mixed with the entering air, and the carbon and hydrogen unite with the oxygen, forming carbon dioxide, CO_2, and water vapor, H_2O, respectively, and give up heat in doing so. If volatile sulphur is present it unites with the oxygen, forming sulphur dioxide, SO_2, and also gives up heat, but its presence is objectionable, as the SO_2, particularly, in the presence of moisture, attacks the metal of the furnace and boiler and causes rapid corrosion. If insufficient

oxygen is present for complete oxidation, the carbon may burn to carbon monoxide, CO, and only a small portion of the available heat be liberated.

3. Combustion of the solid or carbonaceous portion of the fuel. After the hydrocarbons have been driven off and oxidized the remaining solid matter is composed chiefly of carbon and ash. The carbon unites with the oxygen, forming carbon dioxide, carbon monoxide, or both, depending upon the completeness of combustion. The ash, of course, remains unconsumed.

In commercial practice the requirements for perfect combustion are a surplus of air, a thorough mixture of the fuel particles with the air, and a high temperature. The surplus of air above theoretical requirements should be kept to a minimum, but even in the most scientifically designed furnace some excess is essential on account of the difficulty of properly mixing the gases, since the currents of combustible gases and air are apt to be more or less stratified. The products of combustion must be maintained at the kindling temperature until oxidation is complete, otherwise the carbon will be wasted as carbon monoxide or as smoke. The final products of combustion as exhausted by the chimney should consist only of carbon dioxide, water vapor, oxygen, nitrogen, and the oxides of impurities in the fuel.

Prof. Wm. A. Bone of the University, Leeds, England, has recently advocated "flameless incandescent surface combustion" as a means of greatly increasing the general efficiency of industrial furnaces wherever it can be conveniently applied. (Eng. News, Jan. 18, 1912, p. 96; Engineering, April 14, 1911.)

"The distinguishing and essential feature of the new process is that a homogeneous explosive mixture of gas and air, in the proper proportions for complete combustion, is caused to burn without flame in contact with a granular incandescent solid, whereby a large portion of the potential energy of the gas is immediately converted into radiant form." Prof. Bone claims that he has been able to transmit 95 per cent of the available energy of gaseous fuel to the water in specially constructed fire-tube boilers. For a description of the apparatus used see paragraph 55.

When the combustible elements unite with oxygen they do so in definite proportions called the *molecular weights*, which are always the same, and the union produces a fixed quantity of heat. Thus, in the combustion of carbon to CO, 24 pounds of carbon unite with 32 pounds of oxygen, forming 56 pounds of CO; hence one pound of carbon will form

$$\frac{C_2 + O_2}{C_2} = \frac{24 + 32}{24} = 2.34 \text{ pounds of CO.}$$

The heat of combustion will be 4380 B.t.u. per pound of carbon thus consumed.

In the burning to CO_2 one pound of carbon will form

$$\frac{C_2 + 2O_2}{C_2} = \frac{24 + 2 \times 32}{24} = 3\tfrac{2}{3} \text{ pounds of } CO_2 \text{ and liberates 14,540 B.t.u.}$$

Similarly, in burning to H_2O one pound of hydrogen will form

$$\frac{2\,H_2 + O_2}{2\,H_2} = \frac{2 \times 2 + 32}{2 \times 2} = 9 \text{ pounds of } H_2O.$$

(The exact figures, based upon the relative molecular weights, as adopted by the International Committee on Atomic Weights, are

$$\frac{2 \times 2.016 + 32}{2 \times 2.016} = 8.94 \text{ pounds.}$$

For all practical engineering purposes the use of the exact values of the molecular weights is an unnecessary refinement and the decimal factors may well be omitted. In the ensuing calculations only the approximate values will be considered.) If the products of combustion are condensed and their temperature lowered to the initial temperature of the constituent gases the heat liberated will be 61,950 B.t.u. This is known as the *higher heating* value. If the products of combustion are not condensed, which is the usual case in practice, the latent heat of vaporization of the water vapor is not available. The difference between the higher heating value and the unavailable heat is called the *net*, or *lower heating value.* The unavailable portion of the heat depends upon the temperature at which the products of combustion are discharged. This varies with practically every installation. Thus, one pound of water vapor escaping uncondensed in the products of combustion at temperature t_1 degrees F., will carry away approximately $(1090.7 + 0.455\,t_1 - t)$ B.t.u. above initial temperature t degrees F. of the constituent gases. (Carpenter & Diederichs, Exp. Engng., 1911, p. 467.) Since one pound of hydrogen burns to approximately 9 pounds of water vapor, the lower heating value h' will be

$$h' = 61{,}950 - 9\,(1090.7 + 0.455\,t_1 - t) \text{ B.t.u.,} \qquad (0)$$

Many attempts have been made to adopt a standard lower heating value, but the results have been far from harmonious. In a letter dated Jan. 12, 1912, and addressed to the author, the *U. S. Bureau of Standards* recommends "*that the quantity to be subtracted from the gross value to give the net value be taken as the latent heat of vaporization at 0 degrees Centigrade, of the water, formed during combustion, and of that contained in the fuel.*"

This would give the net or lower heat value of hydrogen as

$$61{,}950 - 9\,(1073.4) = 52{,}290 \text{ B.t.u.}$$

For $t_1 = t = 0°$ C. = 32° F., formula (0) gives the same result.

In the ordinary furnace the oxygen is obtained from the atmosphere which, neglecting moisture and a few minor elements, contains the following, mechanically mixed:

	By Volume.	By Weight.
Nitrogen..............	79.04	76.80
Oxygen................	20.96	23.20

For most engineering purposes this relationship may be expressed:

	By Volume.	By Weight.
Nitrogen	79	77
Oxygen	21	23

Hence, in the combustion of one pound of pure carbon the products of combustion contain not only $3\frac{2}{3}$ pounds of CO_2, but $\frac{77}{23} \times 2\frac{2}{3} = 8.92$ pounds of nitrogen, giving a total of $3\frac{2}{3} + 8.92 = 12.58$ pounds. The nitrogen performs no useful office in combustion and is supposed to pass through the furnace without change. It dilutes the products of combustion and reduces the temperature.

Table 8 gives the physical and chemical properties of the substances most commonly met with in connection with combustion.

19. Calorific Value of Coal. — The heat liberated by the combustion of unit weight of fuel is called the calorific value of the fuel. The most rational way of determining the heat of combustion is to burn a weighed sample of coal in an atmosphere of oxygen in a suitable calorimeter. An alternative method is to calculate the heat of combustion from the chemical analysis. An analysis which determines the per cent of fixed carbon, volatile matter, moisture, and ash, is called the *proximate analysis*, while one which reduces the fuel to its elementary constituents of carbon, hydrogen, nitrogen, sulphur, moisture, and ash is called the *ultimate analysis*. The proximate analysis is comparatively easy to make and gives the general characteristics of a fuel. It is made by subjecting the sample to a moderate temperature to expel the moisture, then to a higher temperature until the volatile matter is driven off, and finally to a very high temperature which drives off all carbon as carbon dioxide and leaves the ash as a residue.

By weighing the residue at the end of each operation the various percentages may be computed. For method of making proximate analysis, see "Report of Committee on Coal Testing," Journal of the American Chemical Society, Vol. 21, p. 1116. For method of making both proximate and ultimate analyses, see "Report of Coal Testing Plant," U. S. Geological Survey, No. 48, Part II, 1906, and "Method of Analyzing Coal and Coke," U. S. Bureau of Mines, Technical Paper, No. 8, 1912.

TABLE 8.

DATA RELATIVE TO ELEMENTS MOST COMMONLY MET WITH IN CONNECTION WITH COMBUSTION OF FUEL.

Substance.	Molecular Formula.	Relative Molecular Weight, Oxygen =32.	Chemical Reactions.	Weight per Pound of Substance in First Column.	
				Oxygen.	Air.
Acetylene	C_2H_2	26.02	$2\,C_2H_2+5\,O_2=4\,CO_2+2\,H_2O$	3.08	13.35
Air	....	....	....	......	......
Ash	....	....	....	......	......
Carbon	C_2	24.0	$2\,C+O_2=2\,CO$	1.33	5.78
Carbon	C_2	24.0	$2\,C+2\,O_2=2\,CO_2$	2.66	11.58
Carbon dioxide	CO_2	44.0	....	......	......
Carbon monoxide	CO	28.0	$2\,CO+O_2=2\,CO_2$	0.57	2.47
Hydrogen	H_2	2.016	$2\,H_2+O_2=2\,H_2O$	8.0	34.8
Marsh gas	CH_4	16.03	$CH_4+2\,O_2=CO_2+2\,H_2O$	4.0	17.4
Nitrogen	N_2	28.02	....	......	......
Olefiant gas	C_2H_4	28.03	$C_2H_4+3\,O_2=2\,CO_2+2\,H_2O$	3.43	14.9
Oxygen	O_2	32.0	....	......	......
Sulphur	S_2	32.07	$S_2+2\,O_2=2\,SO_2$	1.0	4.32
Water vapor	H_2O	18.02	....	......	......

Substance.	Mean Specific Heat.	Density and Specific Volume at 32° F., and 14.7 Lbs. per Sq. In.*		Heat of Combustion (Higher Heat Value) B.t.u.†	
		Weight per Cu. Foot.	Cu. Feet per Lb.	Per Pound.	Per Cu. Foot at 32° F., and 14.7 Lbs.
Acetylene	See Par. 21 and Fig. 7.	0.0725	13.79	21,430	1582
Air		0.0807	12.39	....	....
Ash		....	....	....	....
Carbon		145 (solid)	....	4,380	....
Carbon		145 (solid)	....	14,540	....
Carbon dioxide		0.1227	8.15	....	....
Carbon monoxide		0.0781	12.80	4,380	342
Hydrogen		0.0056	177.9	61,950	345
Marsh gas		0.0447	22.37	23,840	1067
Nitrogen		0.0783	12.77	....	....
Olefiant gas		0.0781	12.80	21,450	1685
Oxygen		0.0892	11.21	....	....
Sulphur		125 (solid)	....	4,020	....
Water vapor		....	....	....	....

* Smithsonian tables. † Carpenter and Diederichs, Exp. Eng., 1911.

The formula most commonly used in calculating the heating value of a fuel is based on the ultimate analysis and is known as *Dulong's formula.* This is based on the assumption that the oxygen in the fuel and enough hydrogen to unite with it may be considered inert and the remainder of the hydrogen and all the carbon and sulphur may be treated as free elements, thus:

$$h_d = 14{,}540\,C + 61{,}950\left(H - \frac{O}{8}\right) + 4020\,S,^* \tag{1}$$

in which

h_d = heating value in B.t.u. per pound of fuel.

C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur, respectively, in the fuel.

With fuels low in volatile matter values calculated by means of Dulong's formula agree closely with calorimetric determinations, but may be in considerable error for fuels having more than 20 per cent of volatile matter.

For a comparison between the actual heat values and those calculated by means of Dulong's formula for various coals, see Tables 1 to 5.

The following modification of Dulong's formula, as developed by Mahler, gives results for Pennsylvania and Ohio coals agreeing within 2 per cent of calorimeter determinations. (Trans. A.I.M.E., Feb., 1897.)

$$h_m = 200\,C + 675\,H - 5400, \tag{2}$$

in which C and H are in per cent.

The heating value of certain classes of coals may be estimated from the proximate analysis. Thus, for Illinois coals with ash content under 10 per cent, R. W. Hunt & Co. deduced the formula

$$h_h = 14{,}544\,C + 16{,}515\,V - 10{,}000\,A, \tag{3}$$

in which

h_h = B.t.u. per pound of coal.

C, V, and A = the proportional content of fixed carbon, volatile matter, and ash.

When ash lies between 10 and 15 per cent, the formula will be more accurate if written

$$h_h = 14{,}544\,C + 16{,}515\,V + 354\,A - 1635. \tag{4}$$

* Dulong's formula is usually stated:

(*a*) Heating value per pound = $14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right) + 4000\,S.$

The U. S. Geological Survey uses:

(*b*) Heating value per pound = $14{,}544\,C + 62{,}028\left(H - \frac{O}{8}\right) + 4050\,S.$

Kent ("Steam Boiler Economy," First Edition, p. 47) deduced from Mahler's tests of European coals the following relationship between the approximate heating value and the percentage of fixed carbon in the combustible:

Percentage of Fixed Carbon in Coal, Dry and Free from Ash.	Heating Value B.t.u. per Pound of Combustible.	Percentage of Fixed Carbon in Coal, Dry and Free from Ash.	Heating Value B.t.u. per Pound of Combustible.
100	14,600	68	15,480
97	14,940	63	15,120
94	15,210	60	14,580
90	15,480	57	14,040
87	15,660	55	13,320
80	15,840	53	12,600
72	15,660	51	12,240

Goutal (Comptes Rendus de l'Academie des Sciences, Vol. 135, p. 477) gives carbon a fixed value and considers the heat value of the volatile matter a function of its percentage referred to combustible:

$$h_g = 14{,}760\,C + aV, \tag{5}$$

in which

h_g = B.t.u. per pound of *coal.*
C = proportional content of fixed carbon in the *coal.*
V = proportional content of volatile matter in the *coal.*
a = coefficient as per following table:

$\frac{V}{V+C}$.	a.	$\frac{V}{V+C}$.	a.
05	26,100	.26	18,360
10	23,400	28	17,980
12	22,350	30	17,640
14	21,450	32	17,300
16	20,750	34	17,000
18	20,220	36	16,680
20	19,620	37	16,180
22	19,220	38	15,300
24	18,790	40	14,400

Kent's and Goutal's methods give results which are accurate enough for ordinary work when applied to eastern coals within their range, but they apply with less accuracy to the coals of the middle west and are quite unreliable for coals in the far west and north.

Calorimetric determinations are necessary in all cases where accuracy is required.

Tables 1 to 4 give the proximate and ultimate analyses and the calorimetric and calculated heat values for a number of American coals.

For a means of determining the hydrogen, nitrogen, total carbon and oxygen content of coal from the proximate analysis, see "Ultimate Analysis of Coal," by Prof. L. S. Marks, Power, Dec., 1908, p. 928.

Fuel Calorimeters: See paragraph 458.

Calorific Value of Fuels: Engr., London, Feb. 17, 1911; U. S. Geological Survey, Bulletin Nos. 261, 290, 323, 325, 332; Jour. Franklin Inst., P. Mahler, Jan., 1905; Prac. Engr., U. S., Jan., 1910.

Recent Progress in Calorimetry: Met. and Chem. Engrg., Sept., 1911; Comparison of Calorimeters, Jour. Soc. Chem. Ind., 22–1230, 23,704.

20. Air Required for Combustion. — One pound of carbon in burning to CO_2 requires 2.66 pounds of oxygen or $2.66 \div 0.23 = 11.58$ pounds of dry air. It may be shown in a similar manner that one pound of hydrogen requires 34.8 pounds of dry air. Since the combustible portion of all commercial fuels consists chiefly of carbon and hydrogen the theoretical air requirements may be approximated from the ultimate analysis as follows:

$$A_1 = 11.58\,C + 34.8\left(H - \frac{O}{8}\right), \tag{6}$$

in which

A_1 = weight of dry air required per pound of fuel, pounds.

C, H and O = proportional part of dry weight of carbon, hydrogen and oxygen in the fuel.

$\frac{O}{8}$ = proportional part of the hydrogen supplied with oxygen from the fuel itself.*

Equation (6) is commonly written:

$$A = 35\left(\frac{C}{3} + H - \frac{O}{8}\right). \tag{7}$$

Example: Required the theoretical weight of dry air supplied per pound of coal as fired with analysis as follows:

	Per Cent.		Per Cent.
Carbon	65	Ash and Sulphur	13
Hydrogen	5	Water	8
Oxygen	8	Total	100
Nitrogen	1		

* This term $\left(H - \frac{O}{8}\right)$ does not contain a proper correction for the hydrogen contained in the moisture, for not all of the oxygen in coal is combined with hydrogen. Part of the oxygen is probably combined with nitrogen in organic nitrates and part may be present in carbonates in mineral matter caught in the coal. The error of this assumption, however, lies within the accuracy of the average boiler observations.

Substituting the value of C, H, and O in equation (6)

$$A_1 = 11.58 \times 0.65 + 34.8\left(0.05 - \frac{0.08}{8}\right) = 8.92 \text{ pounds,}$$

the theoretical weight of dry air necessary to burn one pound of coal *as fired.*

Since the coal contains 8 per cent of moisture the weight of dry air required per pound of *dry coal* is

$$\frac{8.92}{0.92} = 9.69 \text{ pounds.}$$

The water and ash only are treated as incombustible, therefore the air required per pound of *combustible* is

$$\frac{8.92}{0.79} = 11.29 \text{ pounds.}$$

Example: Required the character and amount of the products of combustion if one pound of coal, as per following analysis, is completely burned with the theoretical air requirements.

	Per Cent.		Per Cent.
Carbon	65	Ash	12
Hydrogen	5	Water	8
Oxygen	8	Sulphur	1
Nitrogen	1	Total	100

The products of combustion will consist of CO_2, N_2, H_2O, ash, and possible SO_2 or SO_3, thus:

The carbon will produce	$0.65 \times \frac{44}{12} =$	2.38 lbs. of CO_2
The air for the carbon will liberate	$0.65 \times \frac{32}{12} \times \frac{77}{23} =$	5.80 lbs. of N_2
The available hydrogen will produce	$\left(0.05 - \frac{0.08}{8}\right)9 =$	0.36 lbs. of H_2O
The air for the hydrogen will liberate	$\left(0.05 - \frac{0.08}{8}\right)8 \times \frac{77}{23} =$	1.07 lbs. of N_2
The oxygen and inert portion of the hydrogen will appear as vapor	$0.08 + \frac{0.08}{8} =$	0.09 lbs. of H_2O
The nitrogen in the fuel is considered inert *		0.01 lbs. of N_2
The moisture will appear as vapor		0.08 lbs. of H_2O
The sulphur † is usually treated as ash	$0.12 + 0.01 =$	0.13 lbs. of ash
Total products of combustion	$=$	9.92 lbs.

* This is not true since a large percentage of the nitrogen content of the fuel appears in the flue gas in combination with other elements, but the amount is so small compared with that supplied in the air that no appreciable error arises from the assumption that it remains inert and passes through the furnace without change.

† The sulphur content is ordinarily so small that no attempt is made to separate the volatile and non-volatile constituents and the whole is treated as ash. If the

The distribution of the elementary gases and compounds is as follows:

	C_2	H_2	O_2	N_2	Air.	CO_2	H_2O	Ash.
C to CO_2	0.65	......	1.73	5.80	7.53	2.38	......	......
Available H to H_2O	......	0.04	0.32	1.07	1.39	......	0.36	......
O and inert H to H_2O	......	0.01	0.08	......	......	......	0.09	......
Nitrogen in fuel	......	......	......	0.01	......	......	......	......
Moisture in fuel	......	......	......	......	......	......	0.08	......
Ash	......	......	......	......	......	......	......	0.13
Total	0.65	0.05	2.13	6.88	8.92	2.38	0.53	0.13

The weight of gaseous products is $9.92 - 0.13 = 9.79$ pounds, or from the table,

$$0.65 + 0.05 + 2.13 + 6.88 + 0.08 = 9.79 \text{ pounds.}$$

The weight of *dry* gases is

$$9.79 - (0.36 + 0.09 + 0.08) = 9.26 \text{ pounds.}$$

The weight of dry air supplied is, from the table, 8.92, which checks with the results as calculated from equation (6).

In practice the amount of air supplied is measured directly in situations where such measurements can be readily made, or, as is usually the case, it is calculated from the flue-gas analysis.

Air excess is essential in the commercial combustion of solid fuels, and the gaseous products of combustion will contain O_2 and possible CO in addition to CO_2, N_2, H_2O, and SO_2, as obtained from perfect combustion with theoretical air supply.

Example: Required the amount of dry air supplied per pound of coal, as per preceding analysis, if the dry flue gas resulting from the combustion is composed of

CO_2, 14 per cent by volume.
CO, 0.5 per cent by volume.
O_2, 6.0 per cent by volume.
N_2, 79.5 per cent by volume.

Temperature of sample, 62 degrees F., barometer 30 inches.

The weights may be determined from the density given in Table 8.

	Volume ×	Density =	Weight.
CO_2	14	0.1159	1.6226
O_2	6	0.0843	0.5058
CO	0.5	0.0737	0.0368

volatile portion is to be considered the influence of the SO_2 or SO_3 in the flue gas should be included in the heat balance. Some engineers treat one-half the sulphur as volatile and the balance as ash.

These weights may be subdivided into those of their constituents, thus the CO_2 contains $\frac{3}{11}$ of carbon and $\frac{8}{11}$ of oxygen, and the CO, $\frac{3}{7}$ of carbon and $\frac{4}{7}$ of oxygen.

$$\begin{array}{rlcrl} \frac{8}{11} \times 1.6226 = & 1.1800 & \qquad & \frac{3}{11} \times 1.6226 = & 0.4426 \\ \frac{4}{7} \times 0.0368 = & 0.0210 & & \frac{3}{7} \times 0.0368 = & 0.0158 \\ \text{Free oxygen} = & 0.5058 & & & \\ \hline \text{Pounds of oxygen} & 1.7068 & & \text{Pounds of carbon} & 0.4584 \end{array}$$

Weight of oxygen per pound of carbon, $\frac{1.7068}{0.4584} = 3.72.$

Since 23 per cent of air by weight is oxygen, weight of dry air per pound of carbon $= \frac{3.72}{0.23} = 16.2.$

Since the coal used contains 65 per cent carbon, the weight of dry air supplied for the carbon is

$$16.2 \times 0.65 = 10.52 \text{ pounds.}$$

The hydrogen in the fuel required,

$$34.8\left(0.05 - \frac{0.08}{8}\right) = 1.39 \text{ pounds.}$$

The total weight of dry air supplied per pound of coal as fired is

$$10.52 + 1.39 = 11.91 \text{ pounds.}$$

The total weight of the dry products of combustion per pound of coal as fired is

$$10.52 + \frac{77}{100} \times 1.39 + 0.65 + 0.01 = 12.25 \text{ pounds.}$$

It should be noted here that the weight of air *theoretically* required to burn the hydrogen has been added to the weight *actually* required to burn the carbon as indicated by the flue-gas analysis. While this is not exactly correct the error is within the accuracy of the average flue-gas analysis.

It has been previously shown that the coal in question requires 8.92 pounds of air for theoretical combustion, hence the percentage of air excess is

$$100\,\frac{11.91 - 8.92}{8.92} = 33.5 \text{ per cent.}$$

The following method, though perhaps not as readily understood as the one involving the densities of the several gases, is more expeditious. It is based on the principle that the weight of an elementary gas (as O_2 or N_2), referred to hydrogen as unity, is proportional to its *atomic*

weight, and that the vapor density of a compound gas (as CO_2 or CO) is proportional to one-half its *molecular* weight.

Thus the vapor density of $CO_2 = \frac{1}{2}(12 + 2 \times 16) = 22$,
and the vapor density of $CO = \frac{1}{2}(12 + 16) = 14$.

For the example given above:

Weight of CO_2 referred to H as unity	=	$14 \times 22 =$	308
Weight of CO referred to H as unity	=	$0.5 \times 14 =$	7
Weight of O_2 referred to H as unity	=	$6 \times 16 =$	96
		Total	411

Since CO_2 consists of $\frac{3}{11}$ of carbon, and CO, $\frac{3}{7}$ of carbon,

Weight of carbon in CO_2	=	$\frac{3}{11} \times 308 =$	84
Weight of carbon in CO	=	$\frac{3}{7} \times 7 =$	3
		Total	87

The weight of oxygen is $411 - 87 = 324$.

The weight of oxygen per pound of carbon $= \frac{324}{87} = 3.72$ pounds.

The weight of air per pound of carbon $= 3.72 \div 0.23 = 16.2$ pounds, as found by the previous calculations.

The above method may be expressed algebraically,

$$A_2 = 5.8 \frac{2(CO_2 + O) + CO}{CO_2 + CO}, \tag{8}$$

in which

A_2 = weight of dry air per pound of carbon.

CO_2, CO, and O = percentages by volume of the carbon dioxide, carbon monoxide, and oxygen in the flue gas.

The following formula is based upon the same principle as equation (8). (Kent, "Steam Boiler Economy," First Edition, p. 33.)

$$A_3 = \frac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)}, \tag{9}$$

in which

A_3 = weight of dry gas per pound of carbon, other notations as in (8).

The 7 N in equation (9) represents the N supplied by the air. Since the N supplied by the air is 77 per cent of the weight of the air, we have

$$A_4 = \frac{7\,N}{3\,(CO_2 + CO)} \div 0.77 = \frac{3.04\,N}{CO_2 + CO}, \tag{10}$$

in which

A_4 = the weight of dry air supplied per pound of carbon.

N, CO, CO_2 = percentages by volume of nitrogen, carbon monoxide, and carbon dioxide in the flue gas.

The excess of air supplied per pound of fuel may be conveniently determined from the relationship,

$$\frac{\text{Air actually required}}{\text{Air theoretically supplied}} = \frac{N}{N - 3.782\,O}. \qquad (11)$$

N and O are respectively, by volume, the proportional parts of the nitrogen and oxygen in the flue gas. The free oxygen is due to the air supplied and not used. This oxygen was accompanied by 3.782 times its volume of nitrogen. (N − 3.782 O) represents the nitrogen content in the air actually required for combustion. Hence, N ÷ (N − 3.782 O) is the ratio of the air supplied to that required.

The various methods discussed above for determining the weight of air supplied give practically identical results for pure carbon, but vary considerably for fuels containing hydrogen. For fuels containing hydrogen and combustible elements other than carbon the amount of air necessary to oxidize them should be added to the carbon requirements in determining the total air supply.

TABLE 9.

RATIO OF TOTAL AIR SUPPLIED TO THAT THEORETICALLY REQUIRED FOR VARIOUS ANALYSES OF FLUE GASES.

$$\text{Ratio} = \frac{N}{N - 3.782\,O}.$$

CO_2+CO.	N=79. CO_2+CO +O=21.	N=79.5. CO_2+CO +O=20.5.	N=80. CO_2+CO +O=20.	N=80.5. CO_2+CO +O=19.5.	N=81. CO_2+CO +O=19.	N=81.5. CO_2+CO +O=18.5.	N=82. CO_2+CO +O=18.
21	1.02	....	....	....	....	....	....
20	1.05	1.00	1.00	....	....	....	....
19	1.11	1.08	1.05	1.02	1.00	....	....
18	1.17	1.14	1.10	1.08	1.05	1.02	1.00
17	1.24	1.20	1.17	1.13	1.10	1.07	1.05
16	1.32	1.27	1.23	1.20	1.16	1.13	1.10
15	1.40	1.35	1.31	1.27	1.23	1.19	1.16
14	1.51	1.45	1.39	1.35	1.30	1.26	1.23
13	1.62	1.55	1.50	1.44	1.39	1.34	1.30
12	1.76	1.68	1.61	1.54	1.49	1.43	1.38
11	1.92	1.82	1.74	1.66	1.60	1.53	1.48
10	2.11	2.00	1.90	1.81	1.72	1.65	1.59
9	2.35	2.21	2.08	1.97	1.88	1.79	1.71
8	2.65	2.47	2.31	2.18	2.06	1.95	1.86
7	3.03	2.80	2.59	2.44	2.27	2.14	2.03
6	3.55	3.22	2.96	2.74	2.54	2.38	2.24
5	4.27	3.81	3.44	3.14	2.89	2.68	2.50
4	5.37	4.65	4.11	3.68	3.34	3.05	2.83
3	7.23	5.97	5.10	4.45	3.96	3.56	3.25
2	11.06	8.34	6.71	5.63	4.85	4.27	3.82
1	23.51	13.83	9.83	7.64	6.27	6.12	4.64

The weight of air supplied per pound of carbon in the fuel may be roughly determined by the percentage of CO_2 in the flue gas. Thus

for the complete oxidation of pure carbon without air excess, the resulting flue gases should consist of carbon dioxide and nitrogen only, and in the ratio by volume of 21 to 79; therefore 21 per cent of CO_2 in the flue gas is indicative of complete combustion and theoretical air supply. In other words, the ratio by volume of CO_2 to N after complete combustion is practically the same as the ratio of the oxygen to the nitrogen in the air before combustion. This applies only to the combustion of *pure carbon.* For fuels high in volatile matter the per cent of CO_2 in the flue gas for complete combustion with theoretical air requirements is less than 21 per cent, since the oxygen which combines with the hydrogen to form H_2O does not appear in the sample of flue gas as ordinarily tested; thus for heavy crude oil the corresponding maximum content of CO_2 is approximately 16 per cent.

Table 10 gives the weight of air per pound of carbon for different percentages of CO_2 in the flue gas. For fuels rich in volatile matter these figures may be in considerable error. The per cent of CO_2 in the flue gas, however, is an excellent index to the efficiency of combustion for any fuel. See, also, Table 19.

In coal-burning practice, from 15 to 16 per cent of CO_2 is all that can be expected under the best conditions, with an average between 10 per cent and 12 per cent. Anything less than 10 per cent shows an excessive amount of air supplied. Traveling grates unless carefully operated are apt to show as low as 5 per cent of CO_2.

TABLE 10.

WEIGHT OF AIR PER POUND OF CARBON AS INDICATED BY THE PERCENTAGE OF CO_2 IN THE FLUE GAS.

Per Cent of CO_2.	Pounds of Air.	Per Cent of CO_2.	Pounds of Air.	Per Cent of CO_2.	Pounds of Air.
21	12	14	18	7	36
20	12.6	13	19.4	6	42
19	13.3	12	21	5	50.5
18	14	11	22.9	4	63
17	14.8	10	25.2	3	84
16	15.7	9	28	2	126
15	16.8	8	31.5	1	210

The *Importance of CO_2, as an Index to Combustion and in Connection with Flue Gas Temperature, to Boiler Efficiency:* Trans. A.S.M.E., 32–1215. *Flue Gas Analysis and Calculations:* Power, Aug. 9, 1910; Eng. Review, Aug., 1910. *Real Relation of CO_2 to Chimney Losses:* Power, Dec. 7, 1909, p. 969. *Sampling and Analysis of Furnace Gas:* Power, Aug. 22, 1911, p. 282.

See also paragraphs 452–456.

21. Temperature of Combustion. — The *actual* temperature incident to the combustion of a fuel is most satisfactorily determined by means of a suitable thermometer or pyrometer. The *theoretical* temperature of combustion may be calculated from the simple relationship

$$t_1 = \frac{h}{ws} + t, \tag{12}$$

in which

t_1 = final temperature of the products of combustion, degrees F.
h = low calorific value of the fuel, B.t.u. per pound.
s = mean specific heat of the products of combustion.
w = weight of the products of combustion, pounds per pound of fuel.
t = initial temperature of the fuel and air supply, degrees F.

Thus, in the combustion of one pound of carbon with theoretical air requirements, initial temperature 62 degrees F., the maximum theoretical temperature will be

$$t_1 = \frac{14{,}540}{12.58 \times 0.29} + 62 = 4000 \text{ degrees F. (approx.).}$$

No such temperature has ever been obtained in practice from the combustion of carbon in air. The discrepancy between actual and calculated results is attributed to (1) difficulty of effecting complete combustion with theoretical air supply, (2) radiation losses, (3) error in the assumed value of the mean specific heat at this high temperature, and (4) uncertainty of the proportion of the calorific value of the fuel available, at this high temperature, for increasing the temperature of the products of combustion.

An inspection of equation (12) will show that the greater the weight of the products of combustion for a given weight of fuel, the lower will be the temperature of combustion. Evidently, for maximum temperature the weight of air supplied per pound of fuel should be kept as low as possible, consistent with complete combustion. A perfect union of fuel and air in theoretical proportions is almost impossible, and to insure complete combustion an excess of air is necessary. The influence of air dilution on temperature of combustion is best illustrated by a practical example:

Required the theoretical temperature of combustion of carbon in air if 50 per cent air excess is necessary for complete combustion. Since the complete oxidation of one pound of carbon requires 11.58 pounds of air, the weight of the products of combustion will be $11.58 + 0.5 \times 11.58 + 1 = 18.37$ pounds and the final increase in temperature will be

$$t_1 = \frac{14{,}540}{18.37 \times 0.27} = 3000 \text{ degrees F. (approx.).}$$

The values of the mean specific heat (s = 0.29 and s = 0.27) used in the preceding computations are based upon the investigations of Pier, Holburn and Henning, and Langen, as compiled by Prof. G. B. Upton of Cornell University.* Upton recommends the following equations in this connection:

For N_2 and CO, $s = 0.243 + 0.000019\,t,$ (13)

O_2, $s = 0.216 + 0.000014\,t,$ (14)

H_2, $s = 3.369 + 0.00055\,t,$ (15)

Air, $s = 0.237 + 0.000019\,t,$ (16)

CO_2, $s = 0.2+75\times10^{-6}\,t-21\times10^{-9}\,t^2+2.2\times10^{-12}\,t^3,$ (16a)

H_2O, $s = 0.452+7.4\times10^{-6}\,t+92.6\times10^{-9}\,t^2-20.6\times10^{-12}\,t^3,$ (16b)

in which

s = mean specific heat at constant atmospheric pressure and temperature range 0 degrees C. to t degrees C.

t = maximum temperature, degrees C.

Between 1000 degrees C. and 1500 degrees C. the results are uncertain and dependence can be placed in only the first two significant figures in the decimal. Beyond 1500 degrees C. the results are purely conjectural since experiments have not been made at these high temperatures.

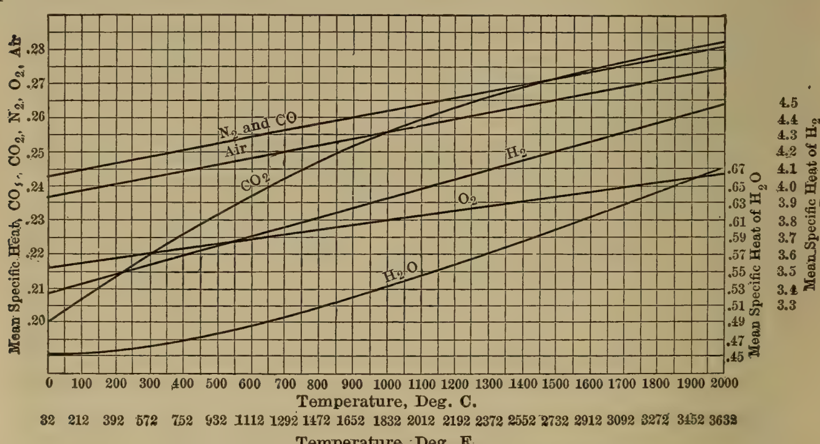


FIG. 7. Mean Specific Heat at Constant Pressure.

The application of the formulas for the mean specific heats at high temperatures to equation (12) necessitates laborious calculations, and since the results are only approximate at the best extreme refinement in calculation is without purpose. The curves in Fig. 7 are plotted

* "Experimental Engineering," Carpenter and Diederichs, 1911, p. 865.

from the above equations, and afford a means of approximating the mean specific heat without the labor of solving the equations.

If the mean specific heats, $s_1, s_2 \ldots s_n$, and weights, $w_1, w_2 \ldots w_n$, of the constituent gases of a compound are known the mean specific heat, s, of the compound may be determined as follows:

$$s = \frac{w_1 s_1 + w_2 s_2 \ldots + w_n s_n}{w_1 + w_2 \ldots + w_n}. \tag{17}$$

The mean specific heat between any two temperatures, t_1 and t, may be determined by substituting $(t_1 + t)$ for t in above equation for the mean specific heat between zero and t degrees.

22. Heat Losses in Burning Coal. — A boiler in order to entirely utilize the heat of combustion of the fuel must be free from radiation and leakage losses, the fuel must be completely oxidized and the products of combustion must be discharged at atmospheric temperature. Commercially such conditions are unobtainable, hence complete utilization of the heat generated is impossible. A boiler which utilizes 83 per cent of the heat value of the fuel is exceptional and an average figure for *very* good practice is not far from 75 per cent. The various losses may be summed up as follows:

1. Loss in the dry chimney gases.
2. Loss due to incomplete combustion.
3. Loss of fuel through the grate.
4. Superheating the hygroscopic moisture in the air.
5. Moisture in the fuel.
6. Loss due to the presence of hydrogen in the fuel.
7. Unburned fuel carried beyond the combustion chamber in the form of soot or smoke.
8. Radiation and minor losses.

Some of these losses are preventable. Others are inherent and cannot be avoided.

23. Loss in the Dry Chimney Gases. — This loss depends upon the type and proportion of the boiler and setting and upon the rate of driving. It is usually the greatest of all the losses. The heat carried away may be expressed:

$$h_1 = W(t_c - t)c. \tag{18}$$

in which

h_1 = B.t.u. lost per pound of fuel,
W = weight of dry chimney gases per pound of fuel.
t_c = temperature of the escaping gases, degrees F.
t = temperature of the air entering the furnace.
c = mean specific heat of the dry gases. (This may be taken as 0.24 for most purposes.)

It will be noted that the magnitude of this loss depends chiefly upon the air dilution and the temperature at which the gases are discharged. Flue temperatures less than 450 degrees F. are seldom experienced

TABLE 11.

HEAT CARRIED AWAY BY THE DRY CHIMNEY GASES PER POUND OF COMBUSTIBLE.

Pounds of Air per Pound of Combustible.	Temperature of Chimney Gases. Deg. Fahr.							
	300°	350°	400°	450°	500°	550°	600°	650°
12 *	750 5.2	905 6.2	1060 7.3	1216 8.7	1370 9.5	1528 10.5	1683 11.6	1840 12.7
15	865 6	1112 7.6	1305 9.1	1498 10.3	1679 11.6	1880 13.0	2072 14.3	2262 15.6
18	1004 7.2	1321 9.1	1550 10.7	1778 12.2	2010 13.9	2235 15.4	2460 17	2692 17.9
21	1266 8.7	1530 10.5	1785 12.3	2060 14.2	2320 16	2582 17.8	2846 19.5	3118 21
24	1440 9.9	1740 12	2040 14	2340 16.1	2640 18.2	2940 20.3	3240 22.4	3540 24.4
27	1611 11.1	1950 13.5	2281 15.7	2620 18.1	2958 20.4	3291 22.7	3628 25	3962 27.4
30	1785 12.4	2160 14.9	2530 17.4	2900 20	3270 22.6	3641 25	4016 27.8	4396 30.4
33	1957 13.5	2362 16.3	2779 19.2	3180 22	3589 24.7	4000 27.6	4405 30.5	4820 33.2
36	2130 14.7	2579 17.8	3020 20.8	3461 23.9	3910 27	4350 30	4798 33	5290 36.6
39	2300 15.9	2781 19.2	3261 22.5	3743 25.8	4220 29.2	4700 32.4	5180 35.7	5670 39
42	2479 17.1	2999 20.6	3508 24.7	4023 27.7	4540 31.3	5052 34.8	5570 39.4	6100 42

* Theoretical requirement.

Large type gives the loss in B.t.u. per pound of combustible.

Small type gives the per cent loss, assuming a calorific value of 14,540 B.t.u. per pound of combustible.

except in connection with economizers, and the air dilution is ordinarily in excess of 50 per cent of theoretical requirements, hence the loss from this cause may range from 8 per cent to 40 per cent of the total heat

generated. In excellent practice it is not far from 12 per cent with a general average of from 20 to 25 per cent. In exceptional cases a loss from this cause as low as 9 per cent has been recorded. (Jour. A.S.M.E., Nov., 1911, p. 1463.)

Table 11 indicates the magnitude of the losses for different chimney temperatures and weights of air per pound of combustible.

24. Loss Due to Incomplete Combustion. — If the volatile gases are not completely oxidized, as when the air supply is insufficient or the mixture of air and gases is not thorough, some of the carbon may escape as CO. Some of the hydrocarbons may also pass through the furnace without being burned. (See Table 12.) The presence of even a small amount of CO in the flue gas is indicative of a very appreciable loss, as will be seen from Table 14. Carbon monoxide is a colorless gas and its presence in the chimney gases cannot be detected by the fireman, consequently the absence of smoke is not an infallible guide for perfect combustion. Since the heat of combustion of C to CO is but 4380 B.t.u. against 14,540 B.t.u. for complete combustion of C to CO_2 this loss may be expressed

$$h_2 = C\,\frac{(14{,}540 - 4380)\,CO}{CO_2 + CO} \tag{19}$$

$$= C\,\frac{10{,}160\,CO}{CO_2 + CO},$$

in which

h_2 = the loss in B.t.u. per pound of fuel,

C = proportional part of carbon in the fuel,

CO_2 and CO are percentages by volume of the flue gases.

TABLE 12.

ANALYSIS OF CHIMNEY GASES.

(Report of Committee for Testing Smoke-preventing Appliances, Manchester, England, 1905.)

Boiler.	Smoky.						Clear.					
	CO_2	O_2	CO	CH_4	H_2	N_2	CO_2	O_2	CO	CH_4	H_2	N_2
No. 1, hand fired	11.00	6.90	0.90	......	......	81.20	......	......	......	......	......	......
	10.65	6.45	2.15	......	......	80.75	......	......	......	......	......	......
No. 1, with smoke-prevention device	......	......	......	......	......	......	7.00	13.50	0	......	......	79.50
							9.00	9.75	0	......	......	81.25
No. 2, hand fired	10.25	8.60	.50	0	0	80.65	......	......	......	......	......	......
No. 3, hand fired	13.25	3.50	.05	0.25	0	82.95	......	......	......	......	......	......
No. 4, fire under caustic pot, hand fired	10.95	1.30	3.00	.70	3.23	80.82	......	......	......	......	......	......
No. 5, split bridge, hand fired	8.75	7.00	3.25	.40	1.00	79.60	......	......	......	......	......	......
No. 6, with smoke-prevention device	......	......	......	......	......	......	7.25	12.00	0	0	0	80.75
No. 7, with smoke-prevention device	......	......	......	......	......	......	7.15	12.15	0	0	0	80.70
No. 8, with smoke-prevention device	......	......	......	......	......	......	8.15	11.10	0	0	0	80.75

TABLE 13.

RELATION OF CO AND COMBUSTION-CHAMBER TEMPERATURES.

(U. S. Geological Survey).

	Per Cent of Black Smoke.						
	0	0 to 10	10 to 20	20 to 30	30 to 40	40 to 50	50 to 60
Number of tests	37	18	56	51	36	17	4
Average per cent of smoke	0	7.1	15.5	24.7	34.7	43.1	52.9
Average per cent of CO in flue gases	0.05	0.11	0.11	0.14	0.21	0.33	0.35
Average per cent unaccounted for in heat balance	9.14	10.60	9.46	10.93	11.41	13.41	13.34
Number of tests *	26	16 *	48	45	32	17	4
Average combustion-chamber temperature (° F.)	2180	2215	2357	2415	2450	2465	2617

* Temperatures in combustion chamber were not determined on all tests.

This loss may be reduced to a negligible quantity in a properly designed and carefully operated furnace. In fact the loss from this cause is often exaggerated and seldom exceeds 2 per cent of the total heat value of the fuel except during the few moments following the replenishing of a burned-down fire with fresh fuel or when the supply of air is checked to meet a sudden reduction in load. In improperly designed furnaces in which the volatile gases are brought into contact with the cooler boiler surface before combustion is complete, the carbon monoxide may

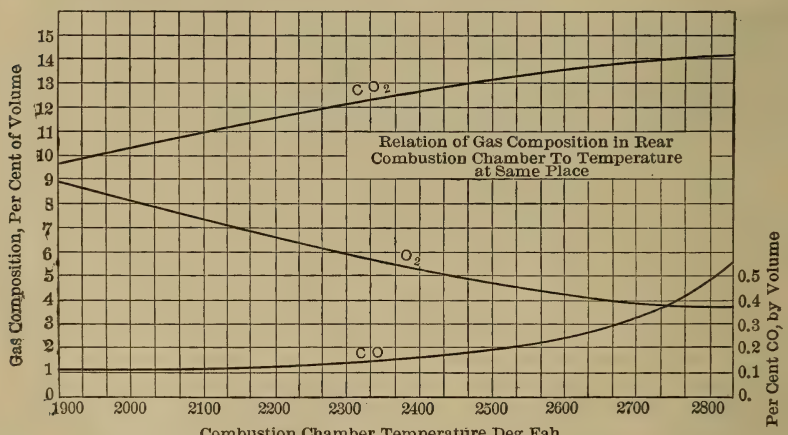


FIG. 8. Relation of Gas Composition in Combustion Chamber to Temperature.

be reduced in temperature below its ignition point and consequently will fail to combine with the oxygen. In such a case the loss may prove to be a serious one. Fig. 8 shows the relation between the composition of the products of combustion in the rear combustion chamber of a 250-horse-power Heine boiler, hand-fired, and the temperature at

the same place. (For an extended discussion of this subject see Jour. West. Soc. Engrs., June, 1907, p. 285.)

25. Loss of Fuel through Grate.— The refuse from a fuel is that portion which falls into the pit in the form of ashes, unburned or partially

TABLE 14.

LOSS DUE TO INCOMPLETE COMBUSTION OF CARBON TO CARBON MONOXIDE.

Per Cent of CO in the Flue Gas by Volume.	Per Cent of CO_2 in the Flue Gas by Volume. 6	8	10	12	14	16
0.2	328 2.2	248 1.7	199 1.3	168 1.1	144 1	126 0.8
0.4	635 4.3	484 3.3	390 2.6	327 2.2	282 1.9	248 1.7
0.6	925 6.3	709 4.8	575 3.9	474 3.2	417 2.8	367 2.5
0.8	1192 8.1	923 6.3	750 5.1	635 4.3	549 3.7	495 3.4
1.0	1494 10.2	1128 7.7	923 6.3	780 5.3	676 4.6	596 4.1
1.2	1690 11.5	1321 9	1085 7.4	923 6.3	801 5.4	708 4.8
1.4	1920 13.1	1512 10.3	1248 8.5	1061 7.2	924 6.3	819 5.6
1.6	2104 14.3	1693 11.5	1400 9.5	1193 8.1	1040 7.1	924 6 3
1.8	2340 16	1865 12.7	1549 10.5	1321 9.0	1151 7.8	1025 7
2.0	2537 17.2	2030 13.8	1690 11.5	1450 9.9	1270 8.6	1129 7.7

Large type gives the loss in B.t.u. per pound of carbon. Small type gives the per cent loss, assuming a calorific value of 14,540 B.t.u. per pound of carbon.

burned fuel, and cinders. The loss from this cause depends upon the size of the fuel, the width of opening in the grate bars, and the type of grate. Coal which necessitates frequent slicing is apt to give greater loss than a free-burning coal. Under good conditions of operation it ought not to exceed 4 per cent of the total heat value of the fuel. In

traveling grates in which a large percentage of the fine fuel falls through the front end of the grate a special hopper is ordinarily installed in the ash pit which reclaims most of it. (See Fig. 102.)

If h_c = calorific value of combustible in the dry refuse,
y = percentage of combustible in the dry refuse,
a = percentage of ash in the coal as fired,
h_3 = heat loss in the refuse, B.t.u. per pound of coal as fired,

$$h_3 = \frac{h_c}{100}\left(\frac{ya}{100 - y}\right). \tag{20}$$

Since the heat balance is only an approximation at the best the calorific value of the combustible in the refuse may be taken as that of the combustible fired.

26. Superheating the Moisture in the Air. — The loss due to this cause is a minor one, though on hot, humid days it may be appreciable. This loss may be expressed

$$h_4 = Mc\,(t_c - t), \tag{21}$$

in which

h_4 = B.t.u. lost per pound of fuel,
M = weight of moisture introduced with the air per pound of fuel,
c = mean specific heat of water vapor, t to t_c degrees F.,
t = temperature of air entering the furnace, degrees F.,
t_c = temperature of chimney gases, degrees F.,

$$M = zwvA, \tag{22}$$

in which

z = relative humidity (see paragraph 271),
w = weight of 1 cubic foot of water vapor at t degrees F. (this may be taken directly from steam tables),
v = volume of 1 pound of dry air at t degrees F., cubic feet,
A = weight of dry air supplied per pound of fuel.

27. Moisture in the Fuel. — Moisture in the fuel represents an appreciable loss in economy if present in large quantities, since the heat necessary to evaporate it into superheated steam at chimney temperature is lost. Firemen occasionally wet the coal to assist coking or to reduce the dust, but moisture thus added necessarily reduces the theoretical furnace efficiency. Under certain conditions wet coal may give a higher evaporation than dry coal. (See paragraph 102.)

The loss due to evaporating the moisture may be expressed

$$h_5 = W\,(1150.4 - c_1\,(t - 32) + c't_s)^* \tag{23}$$
$$= W\,(1182.4 - t + c't_s), \tag{24}$$

* For all purposes c_1 may be taken as unity and in the following calculations and equations it has been considered as such.

in which

h_5 = B.t.u. lost per pound of fuel,

1150.4 = total heat of one pound of saturated steam above 32 degrees F. at atmospheric pressure,

c_1 = mean specific heat of water, 32 to t degrees F.,

W = weight of moisture per pound of fuel,

c' = mean specific heat of water vapor, 212 to t_c degrees F.,

t = temperature of the fuel,

t_c as in equation (21),

t_s = degree of superheat = $(t_c - 212)$.

Carpenter and Diederichs ("Experimental Engineering," 1911, p. 467) give this heat loss as

$$h_5 = W\,(1090.7 + 0.455\,t_c - t). \tag{25}$$

The difference in results between the two formulas is less than $\frac{3}{10}$ of one per cent for ordinary conditions of practice.

28. Loss due to the Presence of Hydrogen in the Fuel. — The hydrogen in any fuel which is not rendered inert by oxygen burns to water and in so doing liberates 61,950 B.t.u. per pound. All of this heat is not available for producing steam in the boiler, since the water formed by combustion is discharged with the flue gases as superheated steam at chimney temperature. This loss is equal to

$$h_6 = 9\,H\,(1182.4 - t + c't_s), \tag{26}$$

in which h_6 = B.t.u lost per pound of fuel,

H = weight of hydrogen per pound of fuel.

All other notations as in equations (24) and (25).

With anthracite coal this loss is approximately 2.5 per cent of the total heat value of the combustible and with bituminous coal it runs as high as 4.5 per cent.

29. Loss due to Smoke. — Visible smoke consists of carbon in a flocculent state and ash mixed with the products of combustion. It is seldom evident in connection with anthracite coal and is generally associated with bituminous fuel. A smoky chimney does not necessarily indicate an inefficient furnace, since the losses due to visible smoke generation seldom exceed 2 per cent; as a matter of fact, a smoky chimney may be much more economical than one which is smokeless. That is to say, a furnace operating with minimum air supply may cause dense clouds of smoke and still give a higher evaporation than one made smokeless by a very large excess of air. There will be some loss due to carbon monoxide, unburned hydrocarbons and soot in the former case, but this may be more than offset by the excessive losses

caused by the heat carried away in the chimney gases in the latter. The amount of combustible in the soot and cinders deposited on the tubes and in various parts of the setting seldom exceeds two per cent of the calorific value of the fuel.

Smoke has become such a public nuisance, particularly in the larger cities, that special ordinances prohibiting its production have been enacted and violators are subject to heavy fines. Effective enforcement of these ordinances renders smoke production very costly and the problem of smokeless combustion becomes a momentous one.

The subject of smoke prevention and smoke-prevention devices is discussed at some length in Chapter V.

30. Radiation and Minor Losses. — These losses are usually determined by difference. That is, the difference between the heat represented in the steam and the losses just mentioned is charged to radiation, leakage, and unaccounted for. Summing up the various losses we have as a rough approximation

	Excellent Practice. Per Cent.	Good Practice. Per Cent.	Average Practice. Per Cent.	Poor Practice. Per Cent.
Heat given to steam	80	70	60	50
Loss in chimney gases	12	18	24	30
Loss due to carbon burning to CO	0	1	2	3
Loss of fuel through grate	0.5	1	2	3
Loss due to moisture in coal, moisture in air, and hydrogen in fuel	3.0	3	3	3.5
Smoke, soot, etc.	0	0.5	1	1.5
Radiation and minor losses	4.5	6.5	8	9

The above heat-loss distribution refers specifically to boilers in continuous operation. In many situations, particularly in large central stations, a considerable portion of the boiler equipment is held in reserve for peak loads and as a consequence the boilers are in actual operation but a short period during the day. The coal burned in banking the fires in order to hold the boilers in readiness is charged to *standby losses*. These losses include the coal required to start up cold boilers, the heat discharged in "blowing off," the fuel lost in cleaning fires and in shutting down. The magnitude of standby losses depends upon the size and character of the boiler equipment and the condition of operation, and may range from 5 per cent to 15 per cent or more of the total heat generated (yearly basis).

Example: Calculate the various heat losses from the following data:

Heat absorbed by the boiler, 76 per cent of the calorific value of the coal as fired.

Analysis of coal as fired:

	Per Cent.		Per Cent.
Carbon	65	Ash and sulphur	13
Oxygen	8	Water	8
Hydrogen	5	Nitrogen	1

Calorific value as fired, 11,850 B.t.u.

Flue-gas analysis:

	Per Cent.		Per cent.
CO_2	14	CO	0.5
O_2	6	N	79.5 (by difference).

Temperature of air entering furnace, 70 degrees F.; temperature of flue gases, 470 degrees F.; temperature of the steam in the boiler, 340 degrees F.; relative humidity of air entering furnace, 80 per cent; combustible in the dry refuse, 20 per cent.

The heat distribution may be referred to the coal as fired, dry coal or combustible. In this problem it is referred to the coal as fired.

DISTRIBUTION OF ACTUAL HEAT LOSSES PER POUND OF COAL AS FIRED.

(Calorific Value of Coal as Fired, 11,850 B.t.u.)

	B.t.u.	Per Cent.
0. Heat absorbed by the boiler, $0.76 \times 11{,}850$	9006	76.00
1. Loss in the dry chimney gases, $h_1 = 12.25^* (470 - 70)\, 0.24$ (equation 18)	1176	9.92
2. Loss due to incomplete combustion, $h_2 = 0.65 \dfrac{10{,}160 \times 0.05}{14 + 0.05}$ (equation 19)	227	1.91
3. Loss in combustible in refuse, $h_3 = \dfrac{11{,}850 \div 0.79}{100}\left(\dfrac{20 \times 13}{100 - 20}\right)$ (equation 20)	487	4.11
4. Loss due to moisture in the air, $h_4 = 0.8 \times 0.00115 \times 13.3 \times 11.91^* \times 0.46\, (470 - 70)$ (eq. 22)	27	0.23
5. Loss due to moisture in the coal, $h_5 = 0.08\, [(1182.4 - 70) + 0.46\, (470 - 212)]$ (equation 24)	98	0.83
6. Loss due to hydrogen in the fuel, $h = 9 \times 0.05\, [(1182.4 - 70 + 0.46\, (470 - 212)]$ (eq. 26)	554	4.67
7. Radiation and unaccounted for (by difference)	275	2.33
Total	11,850	100.00

* See third example, paragraph 20.

The *inherent* or *unpreventable* losses may be summarized as follows:

1. Heat absorbed by the theoretical weight of dry chimney gases in being heated from boiler room to boiler steam temperature.

2. Heat required to evaporate and superheat the moisture in the fuel from boiler room to boiler steam temperature.

3. Heat required to evaporate and superheat the H_2O formed by the combustion of hydrogen in the fuel from boiler room to boiler steam temperature.

4. Heat required to superheat the moisture in the air (theoretical requirements) from boiler room to boiler steam temperature.

For the preceding problem:

DISTRIBUTION OF INHERENT HEAT LOSSES PER POUND OF COAL AS FIRED.

	B.t.u.	Per Cent.
1. Inherent loss in the dry chimney gas, $9.26^* \times (340-70)\,0.24$	600.0	5.06
2. Inherent loss due to moisture in coal, $0.08\,[1182.4-70+46\,(340-212)]$	93.7	0.79
3. Inherent loss due to H_2O formed by the combustion of hydrogen, $9\times0.05\,[1182.4-70+0.46\,(340-212)]$	427.0	3.60
4. Inherent loss due to "humidity" of the air, $0.8\times0.00115\times13.3\times8.92^*\times0.46\,(340-70)$	13.5	0.11
5. Heat absorbed by ideal boiler (by difference)	10,715.8	90.44
Total	11,850.0	100.00

* See first example, paragraph 20.

31. Size of Coal — Bituminous. — Coal is usually marketed in different sizes, ranging from *lump coal* to *screenings.* The latter furnish by far the greater part of the stoker fuel used. For maximum efficiency coal should be uniform in size. With hand-fired furnaces there is usually no limit to its fineness and larger sizes can be used than with stokers. As a rule the percentage of ash increases as the size of coal decreases. This is due to the fact that all of the fine foreign matter separated from larger coal, or which comes from the roof or the floor of the mine, naturally finds its way into the smaller coal. The size best adapted for a given case is dependent upon the intensity of draft, kind of stoker or grate, and the method of firing, and its proper selection often affords an opportunity to effect considerable economy. The influence of the size of screenings on the capacity and efficiency of a boiler in a specific case is illustrated in Fig. 9. The curves are plotted from a series of tests conducted with Illinois screenings on a 500-horse-power B. & W. boiler, equipped with chain grates, at the power house of the Chicago Edison Company.

Influence of Thickness of Fire. — See paragraph 82.

Size of Coal: Some Characteristics of Coal as affecting Performances with Steam Boilers: Jour. West. Soc. Engrs., Oct., 1906, p. 528.

32. Washed Coal. — Coal is washed for the purpose of separating from it such impurities as slate, sulphur, bone coal, and ash. All of these impurities show themselves in the ash when the coal is burned. Screenings contain anywhere from 5 per cent to 25 per cent

of ash and from 1 per cent to 4 per cent of sulphur. Washing eliminates about 50 per cent of the ash and some of the sulphur. Table 15 gives some idea of the effects of washing upon a number of grades of coal.

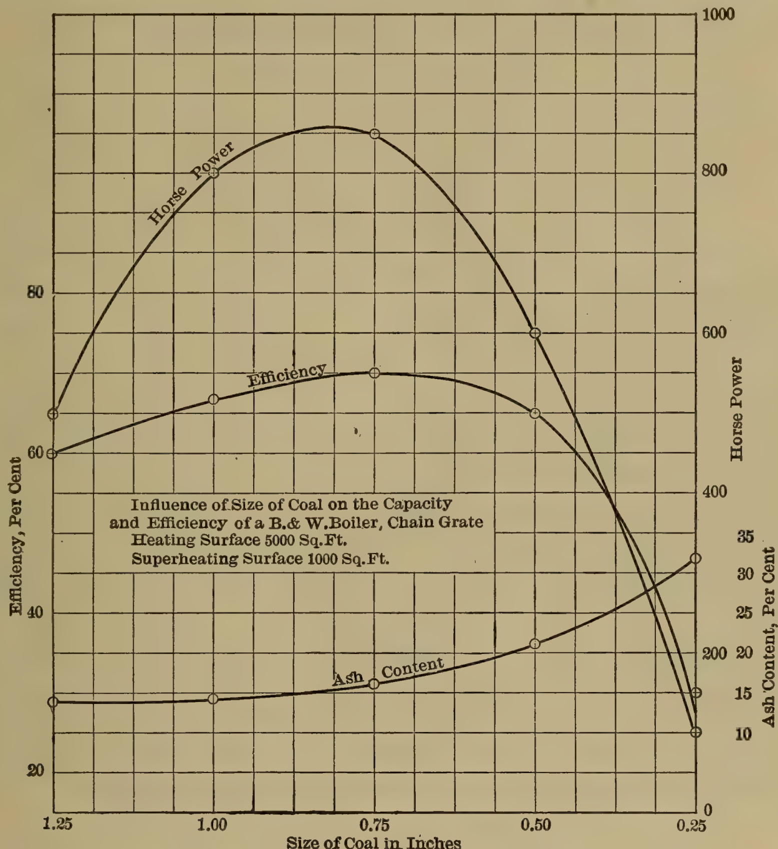


FIG. 9. Influence of Size of Coal on Boiler Capacity and Efficiency.

The evaporative power of the combustible is practically unaffected by washing and the greater part of the water taken up by the coal is removed by thorough drainage. Many coals otherwise worthless as steam coals are rendered marketable by washing. Washed coals are usually graded as follows:

Size.	Screens.	
No. 1	Over $1\frac{3}{4}$	Under $2\frac{1}{2}$
2	$1\frac{1}{8}$	$1\frac{3}{4}$
3	$\frac{3}{4}$	$1\frac{1}{8}$
4	$\frac{1}{4}$	$\frac{3}{4}$
5	...	$\frac{1}{4}$

Numbers 3 and 4 are excellent sizes for use in connection with stokers and No. 5 is well adapted for hand furnaces where smoke prevention is essential.

TABLE 15.

EFFECT OF WASHING ON BITUMINOUS COALS.

(*Journal W.S.E.*, December, 1901.)

	Before Washing. (Per Cent.)			After Washing. (Per Cent.)		
	Ash.	Sulphur.	Fixed Carbon.	Ash.	Sulphur.	Fixed Carbon.
Belt Mountain, Mont.	18.74	3.34	43.72	5.56	2.40	48.39
Wellington Colliery Co., Vancouver Island (new coal)	35.00	......	38.00	8.90	......	56.90
Alexandria Coal Co., Crabtree, Pa.	10.60	1.30	....	6.21	0.61	....
DeSoto, Ill.	18.00	......	44.00	4.20	......	57.00
Northwestern Improvement Co., Roslyn, Wash.	16.30	0.57	45.90	9.70	0.40	47.86
Luhrig Coal Co., Zaleski, Ohio	15.80	1.90	....	8.00	0.87	50.90
Rocky Ford Coal Co., Red Lodge, Mont.	25.30	......	37.80	8.50	......	47.20
Buckeye Coal and Ry. Co., Nelsonville, Ohio	13.77	1.05	49.04	4.30	0.89	54.82
New Ohio Washed Coal Co., Carterville, Ill.	9.48	0.78	55.00	4.85	0.69	63.00

33. Purchasing Coal. — Engineers fail to agree as to the specifications best suited for the purchase of coal. Some extensive purchasers require elaborate analyses and others specify only the size and grade of the fuel. Every essential requirement of the purchaser may be fulfilled by confining them to the four following characteristics:

Moisture.
Ash.
Size of coal.
Calorific value of the coal.

Although moisture is a great and uncertain variable, and the producer can exercise no control over this factor, still the purchaser should protect himself against excessive moisture by stipulating an amount consistent with the average inherent moisture in the coal, and proper penalty should be fixed for delivery in excess of the amount allowed, a corresponding bonus being paid for delivery of less than contract amount. Considerable attention should be given to the percentage of earthy matter contained. The amount of earthy matter usually fixes the heating value of the coal, since the heating value of the combustible

is practically constant. The effect of ash on the heat value of Illinois screenings as fired under a B. & W. boiler with chain grate is shown in Fig. 10. This value varies with the different types of boilers, grates, and furnaces, but is substantially as illustrated. The amount of refuse

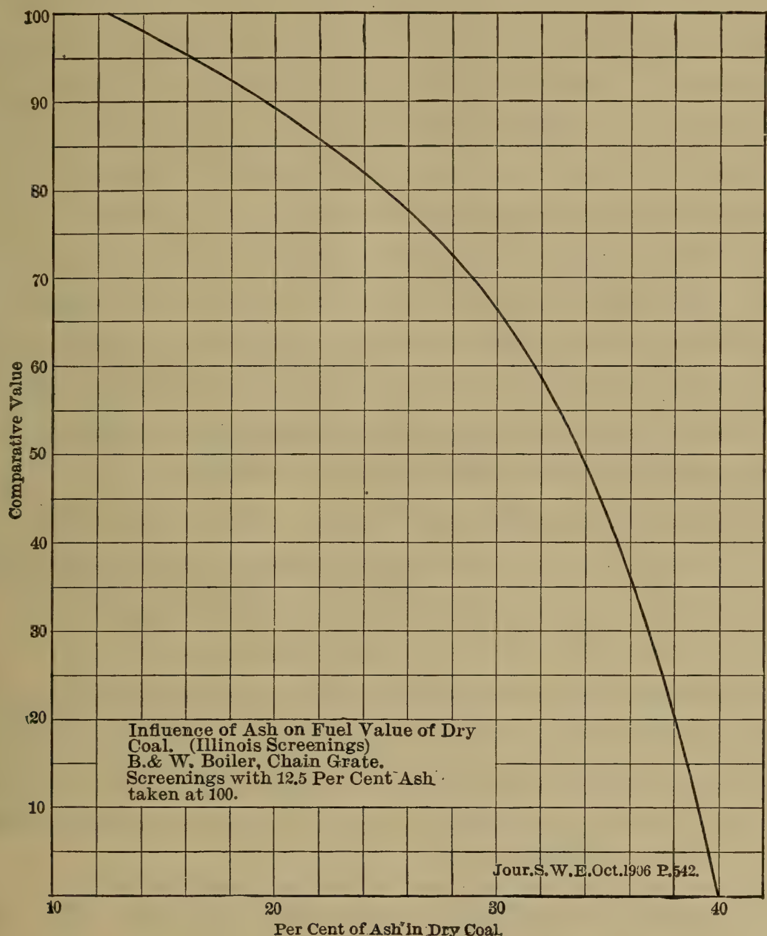


FIG. 10. Influence of Ash on Fuel Value of Dry Coal.

in the ash pit is always in excess of the earthy matter as reported by analysis.

The maximum allowable amount of sulphur is sometimes specified, since some grades of coal high in sulphur cause considerable clinkering. But sulphur is not always an indication of a clinker-producing ash, and a more rational procedure would be to classify a coal as clinker-

ing or non-clinkering according to its behavior in the particular furnace in question, irrespective of the amount of sulphur present. An analysis of the various constituents of the ash is necessary to determine whether or not the sulphur unites with them to produce a fusible slag, and as such analyses are usually out of the question on account of the expense attached they may well be omitted.

The heating value of the coal as determined by a sample burned in an atmosphere of oxygen does not give its commercial evaporative power, since this depends largely upon the composition of the fuel, character of grate, and conditions of operation. It serves, however, as a basis upon which to determine the efficiency of the furnace. In large plants where a number of grades of fuel are available it is customary to conduct a series of tests with the different grades and sizes, and the one which evaporates the most water for a given sum of money, other conditions permitting, is the one usually contracted for. In designing a new plant particular attention should be paid to the performance of similar plants already in operation, and that fuel and stoker should be selected which are found to give the best returns for the money. Where smoke prevention is a necessity the smoke factor greatly influences the choice of fuel and stoker. (See, also, paragraph 480.)

The Purchase of Coal: Eng. Mag., Mar., 1911; Jour. A.S.M.E., Mar., 1911; Power, Apr. 6, 1909, p. 642.

The Purchase of Coal by the Government under Specifications: Bureau of Mines, Bull. No. 11, 1910; U. S. Geol. Survey, Bulletins No. 339, 1908; No. 378, 1909.

The Fusing Temperature of Coal Ash: Power, Nov. 28, 1911, p. 802.

The Clinkering of Coal: Eng. News, Dec. 8, 1910.

34. Powdered Coal. — The value of powdered coal as a fuel for steam boiler plants has long been known, and appliances for pulverizing and feeding the coal have been on the market for a number of years. However, despite the many advantages of powdered fuel and the apparent success of some of the systems of burning it, little progress has been made toward its general adoption.

Some of the advantages obtained in burning powdered coal are:

(*a*) Complete combustion and total absence of smoke. The coal in the form of dry impalpable dust is induced or forced into the zone of combustion, where each minute particle is brought into contact with the necessary amount of air and complete oxidation is effected without the excess of air which accompanies the firing with lump coal, provided the furnace is properly proportioned. With a properly designed setting there is complete absence of smoke.

(*b*) A cheaper grade of bituminous coal may be burned, since the per cent of ash and moisture has little effect on the completeness of

combustion and the full value of the fuel is more nearly realized than with ordinary firing.

(*c*) The plant may be rapidly forced above its rated capacity and sudden demands for power readily met.

(*d*) The labor of firing is reduced to a minimum.

35. Furnaces for Burning Powdered Coal. — In burning ordinary bulk coal the mass of incandescent fuel stores up a quantity of heat to effect the distillation and ignition of the volatile matter in the green fuel. With pulverized coal a refractory lining is necessary to bring about the same result. A large combustion chamber is necessary and the shape of furnace and path of flame must be such as to provide a uniform distribution of heat over the boiler-heating surface without direct impingement of flame. Fire-brick arches and target walls are not to be recommended, owing to the rapid destruction of the brickwork under the intense heat of combustion. Fig. 11 shows a successful arrangement of boiler and furnace for burning coal dust. (For detailed description of this apparatus, see Power, Feb. 14, 1911, p. 264.)

36. Types of Powdered-coal Burners. — Powdered-coal burners may be grouped into two general classes:

1. The dust-feed burner, in which the coal is supplied in the powdered form, and

2. The self-contained burner, in which the coal is crushed, pulverized, and fed to the furnace simultaneously.

The dust may be fed into the furnace by

1. Natural draft,
2. Mechanical means, or by
3. Forced draft.

The following outline gives a classification of a few of the best known coal-dust burners:

Natural Draft	Natural Draft Feed	Pinther Wegener	Dust Feed
	Brush Feed	Schwartzkopff	
Forced Draft	Blower Feed	Cyclone Triumph	
	Compressed Air	Eng. and Powdered Fuel Company	Self-contained
	Paddle Wheel	Ideal Blake	

37. Pinther Apparatus. — Fig. 12 shows a section through a Pinther coal-dust feeder, illustrating the principles of the "natural-draft feed" type. The powdered coal is placed into hopper *B*, from which it is fed by rollers *a*, *a* into the chamber leading to the furnace *C*. The dust

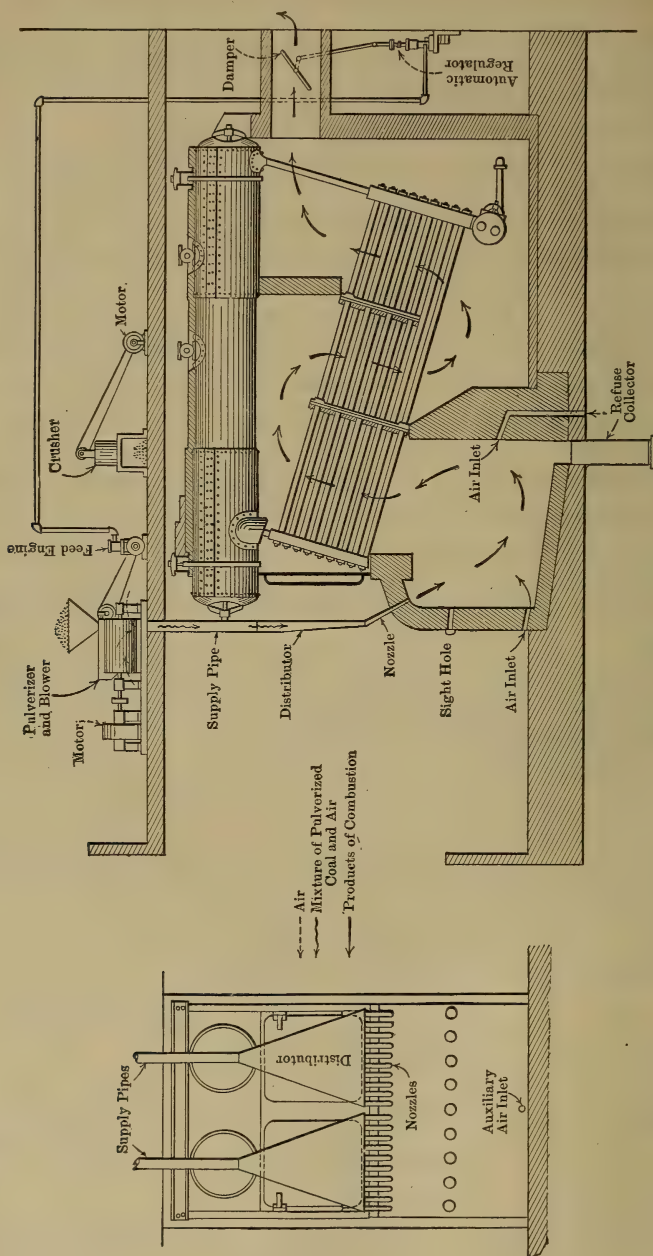


Fig. 11. 300-h.-p. Boiler with Blake Powdered-coal Furnace.

falls in a thin stream and is caught up by the current of air and drawn into the furnace as indicated. The furnace is lined with refractory material heated to a sufficiently high temperature to ignite the fuel and burn it in suspension. The chief drawback to a burner of this type is its limited capacity. Any attempt to feed large quantities of fuel into the furnace necessitates such a strong current of air as to carry the particles of dust beyond the zone of combustion before they are completely consumed. Within the limits of its capacity it is an efficient and simple apparatus, but is open to the same objection as all burners of this type in that it necessitates the storage of powdered coal. This apparatus is not much in evidence in boiler plants.

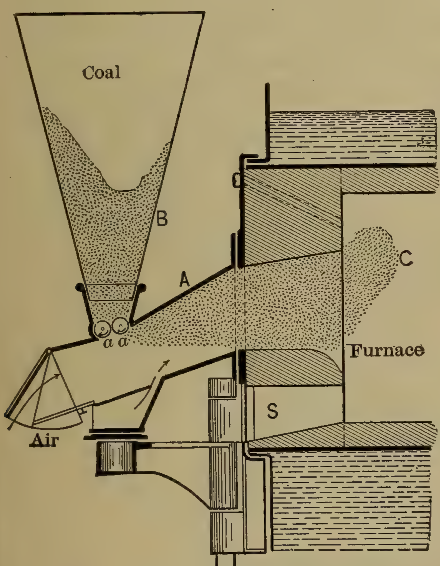


Fig. 12. Pinther Coal-dust Feeder.

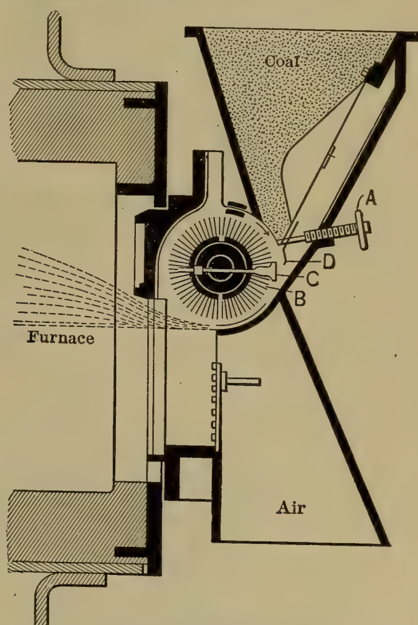


Fig. 13. Schwartzkopff Coal-dust Feeder.

38. Schwartzkopff Apparatus. — Fig. 13 shows a section through a Schwartzkopff feeder, illustrating the principles of the brush-feed, natural-draft system. It is a very simple and practical dust feeder, though open to the objection of all systems which require the coal to be ground and pulverized in separate machines. The fuel is placed in a hopper and its supply to the brush is regulated by the hand screw *A* and the spring plate bottom of the hopper. The brush, consisting of a number of flat steel leaves $\frac{1}{32}$ inch by $\frac{1}{8}$ inch wide, revolves at a high speed, 1000 to 1200 r.p.m. and forces the dust into the furnace. The air for combustion is admitted either through the grates in the ordinary

way or through the lower chamber of the burner. To prevent the dust from bridging in the hopper, a small hammer C is fitted to the brush so that it will strike the plate D and agitate the dust. This apparatus is meeting with much success in connection with annealing furnaces, but is still in the experimental state as far as boiler firing is concerned.

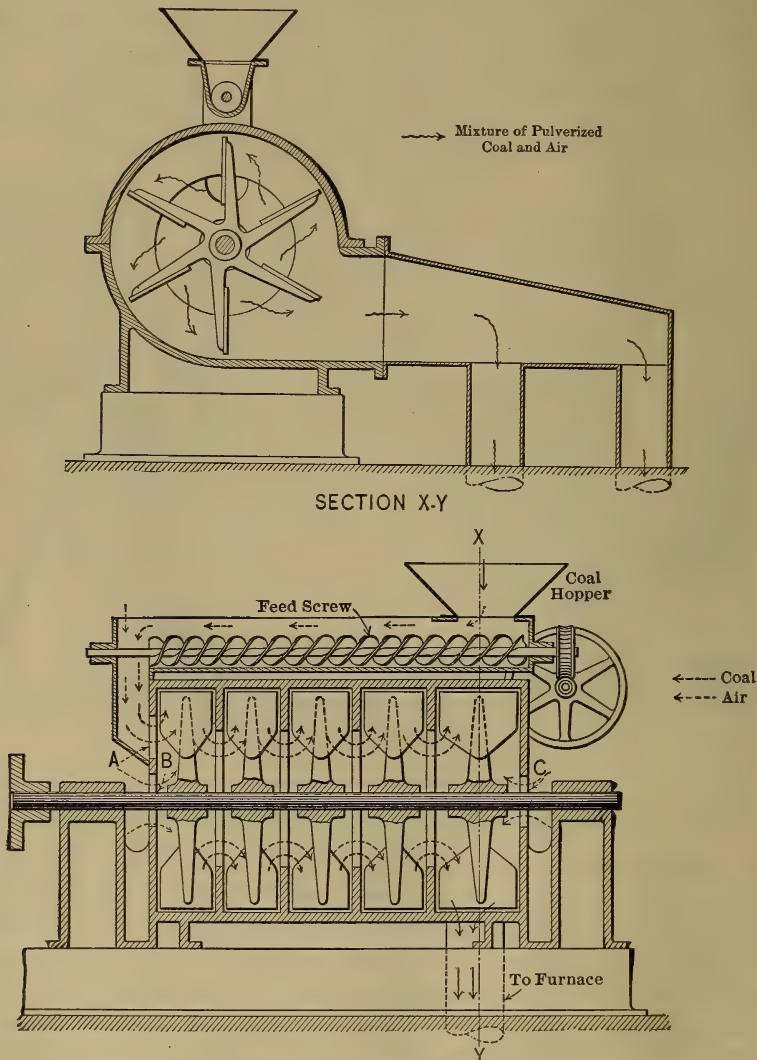


FIG. 14. Blake Coal-dust Feeder.

39. Blake Pulverizer and Feeder. — Fig. 14 gives a sectional view of the Blake apparatus, and is a typical example of a self-contained

system. It comprises a multi-stage centrifugal pulverizer, coal hopper, conveyor and fan mounted on a single bedplate. Referring to the illustration, coal previously crushed to nut size is fed to the hopper from the bottom of which it is conveyed by an endless screw to the first stage of the pulverizer. The lumps are thrown out radially by centrifugal force, due to the rapidly revolving bats, and are reduced to a dust by percussion and attrition. The largest chamber contains a fan, the function of which is to draw the pulverized material successively from one chamber to another and finally deliver it to the discharge spouts. The air is drawn into the fuel chamber with the coal through passage *A*, and also through opening *B* around the shaft. After entering the fan chamber, the mixture of coal dust and air receives an additional supply of air through opening *C*. The apparatus may be belt-driven or direct-connected and runs at about 1200 to 1600 r.p.m., requiring 8 to 12 horse power for its operation. Experience has demonstrated that as much as 14 per cent of moisture in the coal has little effect on the pulverization and burning.

Fig. 11 shows a successful commercial application of the system to a 300-horse-power boiler. Table 16 gives the results of an evaporation test of this installation.

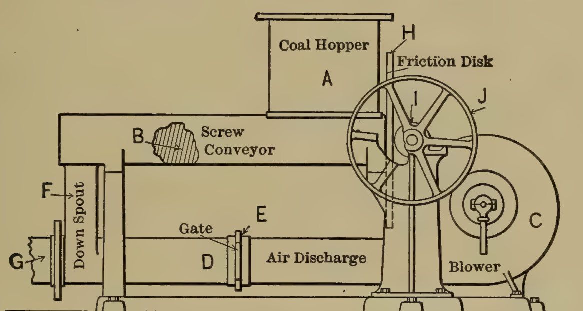


FIG. 15. Triumph Coal-dust Feeder.

40. Triumph Apparatus. — Fig. 15 illustrates the Triumph coal-dust feeder as designed by the C. O. Bartlett & Snow Company, Cleveland, Ohio.

The coal is fed from storage bin to hopper *A* and feed worm *B*. The latter forces it down spout *F* directly to delivery tube *D*, where it is caught by the air draft and fed into the furnace.

The amount of feed depends upon the speed of the feed worm, which is driven by a friction disk *I* against the flange plate *H*. This disk is moved in or out by handle, so as to get any speed desired. The air is furnished by fan *C*, the amount being controlled by valve *E*.

41. Efficiency of Powdered-coal Furnaces. — A comparison of a number of tests of hand-fired and powdered-coal furnaces with different types of feeders shows a decided gain in efficiency of the powdered coal over the hand-fired where the fuel is of a low grade. The gain becomes less marked with fuel of fair quality and disappears entirely with good fuel and properly manipulated automatic stokers. A test made by G. H. Barrus on a 250-horse-power B. & W. boiler at the General Electric Works in connection with a coal-dust feeder manufactured by the Phœnix Investment Company of New York gave a boiler and furnace efficiency of 75.3 per cent. A test of a 135-horse-power return tubular boiler with this same stoker gave a combined efficiency of boiler and furnace of 80 per cent. These figures, however, have been equaled and even exceeded in special hand-fired automatic-stoker tests, and only a comparative test of the two systems under similar conditions will show their respective efficiencies.

Table 16 gives the results of a test of a 300-horse-power B. & W. boiler equipped with the Blake system. This plant represents the best practice in powdered-coal burning at this date (Jan., 1912).

TABLE 16.

TEST WITH PULVERIZED COAL AT THE HENRY PHIPPS POWER PLANT.

Duration of test, hours	6
Total weight of coal fired, pounds	5,160
Total weight of water, pounds	56,160
Average temperature feed water, degrees Fahrenheit	186
Average steam pressure, pounds per square inch	162.3
Factor of evaporation	1.084
Water evaporated per pound of coal	10.88
Water evaporated per pound from and at 212 degrees, pounds	11.725
Boiler efficiency (coal containing 14,350 B.t.u.), per cent	78.93
Horse power of boiler	294.6
Builder's rating	300
Temperature of escaping gases, degrees Fahrenheit	386
Cost of coal, 2.50 tons at $1.315 per ton	$3.392
Cost of coal per pound	0.0006575
Pounds of coal per boiler horse power per hour	2.92
Cost of coal per boiler horse power, cents	0.192

42. Rate of Combustion with Powdered Fuel. —In forcing large quantities of dust into the furnace the velocity imparted to the particles may be so great as to carry them beyond the zone of combustion before oxidation is complete, with the result that the flues and the back of the furnace become covered with unconsumed carbon. So much depends upon the depth of the furnace and the arrangement of the regenerative surface that no specific figures can be given as to the maximum rate of

combustion that can be efficiently effected. At ordinary rates of combustion the small particles of fuel are completely oxidized while in the combustion chamber and there is total absence of smoke. The use of anthracite coal is practically excluded from this type of stoker unless mixed with coal high in volatile matter. This is due to the fact that fixed carbon burns more slowly than the hydrocarbon gases and the temperature of ignition is higher; hence the most gentle draft will carry away the particles before they are completely consumed. With fuels high in volatile matter the hydrocarbons are distilled at a comparatively low temperature, forming an inflammable gas which burns rapidly with the fixed carbon. A mixture of 30-per-cent bituminous and 70-per-cent anthracite has been successfully burned in the powdered form.

43. Draft for Powdered Fuel. — A study of a number of tests of boilers burning powdered coal shows that the necessary draft is very low and ranges from 0.05 to 0.2 inch of water and averages not far from 0.1 inch.

44. Storing Powdered Fuel. — Most cities limit the storage of powdered coal to such a small quantity as to prohibit the use of fuel feeders of the "dust-feed" type in plants of any size not provided with a pulverizing and crushing system. Coal dust mixed with air is often claimed to be of an explosive nature and many accidents are reported to have resulted from this cause. Many engineers, however, refute this on the basis of experiments which show that explosion can only occur at temperatures high enough to drive off the volatile gases.

Explosibility of Coal Dust: Engr., Jan. 27, 1911; Mar. 31, 1911. Bureau of Mines, Bulletin No. 20, 1911. Fuel, Jan. 12, 1909, p. 294.

45. Depreciation of Powdered-coal Furnaces. — To withstand the intense heat of combustion, brickwork of the highest quality is essential, since common fire brick are soon reduced to a liquid slag. A good quality of firebrick will withstand the heat for several months without renewals provided the furnace is properly enclosed, otherwise the strain of expansion and contraction due to alternate heating and cooling will crack the brick. Excellent results have been obtained from the use of bricks composed chiefly of the refuse from a carborundum slag, but the high cost has prevented their general use.

46. Cost of Pulverizing Coal. — In stokers of the "Blake Pulverizer" type in which the grinding and feeding are carried on simultaneously in a self-contained apparatus, the power consumed varies from 2 to 10 per cent of the total power developed, depending upon the nature of the fuel, the load factor, the efficiency of the driving mechanism, and the degree of fineness of the powdered fuel; 5 per cent is a

fair average. The best results are obtained when 95 per cent of the dust will pass a 100 mesh and 75 per cent a 200 mesh, though satisfactory results have been obtained with as low as 40 mesh. Powdered coal in the open market ranges from 50 cents to 75 cents a ton above the price of the same coal in the form of screenings.

Firing Boilers with Pulverized Coal: Power, Feb. 14, 1911.

Pulverized Fuel: Eng. Mag., Jan., 1908; Jour. W. S. Engrs., Feb., 1904.

Use of Pulverized Coal Under Steam Boilers: Power, Apr., 1904, March, 1904; Eng. and Min. Jour., Dec. 16, 1905; Col. Guard., Feb. 16, 1912.

Tests of Pulverized Fuel: Engr. U. S., Apr. 1, 1904; Power, May, 1904, Feb. 14, 1911.

Types of Coal Dust Burners: Engr. U. S., Apr. 1, 1904; Jan. 1, 1903; Power, Mar., 1904.

Burning Low Grade Coal Dust: Power, Sept. 12, 1911, p. 393.

47. Fuel Oil. — The recent development of oil wells in the Western and Gulf States, with the consequent enormous increase in production, has given a marked impulse to the use of crude oil for fuel purposes in steam power plants. Where economic and commercial conditions permit, it is the most desirable substitute for coal. The total absence of smoke and ashes, prompt kindling and extinguishing of fires, extreme rate of combustion, and ease with which it can be handled and controlled are marked advantages in favor of fuel oil. The reduction in volume and weight over an equivalent quantity of coal for equal heating values and the increase in boiler efficiency are factors of no mean importance, particularly in connection with marine or locomotive work. In stationary work the chief objections are the difficulty in securing ample storage capacity and the increased rate of insurance. An objection sometimes raised against oil fuel is the increased depreciation of the setting, but in a well-designed setting this figure is only nominal and of secondary importance. However, in spite of the many advantages presented in the use of fuel oil for power plant purposes, the comparatively limited supply prevents its adoption as a general fuel and limits its use to the plants most favorably located.

48. Chemical and Physical Properties of Fuel Oil. — Crude oil, as pumped at the wells, consists principally of various combinations of hydrogen and carbon, together with small amounts of nitrogen, oxygen, sulphur, water in emulsion and silt. The nitrogen and oxygen may be classified with the moisture and silt as inert impurities. The moisture in oil fuel should not exceed 2 per cent, since it not only acts as an inert impurity, but must be converted into steam in the furnace and thus still further reduces the heat value per pound. The sulphur, though combustible, has a low calorific value and is otherwise undesirable. From Table 17 it will be seen that the physical properties

TABLE 17.

ANALYSES OF TYPICAL AMERICAN FUEL OILS.

Location.	Authority.	Physical Properties. Gravity. Baumé at 60° F.	Physical Properties. Gravity. Specific at 60° F.	Flash Point, Deg. F.	Burning Point, Deg. F.	Viscosity at 68° F., Engler Scale.	Chemical Properties. C	Chemical Properties. H	Chemical Properties. O	Chemical Properties. S	B.t.u. per Lb.
California crude:											
Coalinga	Bulletin No. 19 U. S. Bureau of Mines (1912).	17.52	0.9498	192	230	341.5	86.37	11.30	1.14	0.60	18,727
Kern River		15.16	0.9645	226	266	915.6	86.36	11.27	0.74	0.89	18,553
McKitterick		16.37	0.9566	188	207	200.0	86.51	11.41	0.58	0.74	18,508
Midway		16.34	0.9570	172	210	518.0	86.58	11.61	0.74	0.82	18,613
Sunset		14.37	0.9701	192	235	527.0	85.64	11.37	0.84	1.06	18,478
Kansas crude	B. F. McFarland	31.66	0.866	52	77		85.40	13.07			
Louisiana crude	C. E. Coates									0.34	19,814
Ohio distillate	Deville	27.83	0.887				84.20	13.10	2.70*		18,718
Ohio distillate	N. W. Lord	38.25	0.838	177	212						
Pennsylvania crude	Deville	39.50	0.826				82.00	14.80	3.20*		17,930
Pennsylvania distillate	Deville	27.80	0.886				84.90	13.70	1.40*		19,210
West Virginia crude	Deville	36.46	0.841				84.36	14.10	1.60*		18,400
Wyoming crude	E. E. Slosson	20.00	0.933	273	343					0.67	19,440
Texas crude	Denton	22.17	0.920	142	181		84.60	10.90	2.87*	1.63	19,060
Texas distillate	U. S. Naval Report	21.18	0.926	216	240		83.26	12.41	3.83	0.50	19,481

* O + N.

of oils from different localities in the United States differ widely, while the chemical constituents vary but slightly. For example, the oils given in the table differ greatly in volatility, specific gravity, and viscosity, but have approximately the same percentages of carbon and hydrogen. Taking hydrogen and carbon as the principal constituents it is found that oils rich in hydrogen are lighter in weight than those rich in carbon. Other things being equal, oils rich in hydrogen have a higher calorific value than those rich in carbon, but the heavier oils are usually the cheaper. The relation between heating value and specific gravity for anhydrous California oil is as follows:

TABLE 18.

APPROXIMATE RELATION BETWEEN THE HEATING VALUE AND SPECIFIC GRAVITY.
(Professor Le Conte, University of California.)

Degrees, Baumé.	Specific Gravity.	Weight per Barrel.	B.t.u. per Pound.	B.t.u. per Barrel.	Degrees, Baumé.
10	1.0000	350.035	18,280	6,398,600	10
11	0.9929	347.55	18,340	6,374,100	11
12	0.9859	345.10	18,400	6,349,800	12
13	0.9790	342.68	18,460	6,325,900	13
14	0.9722	340.30	18,520	6,302,400	14
15	0.9655	337.96	18,580	6,279,300	15
16	0.9589	335.65	18,640	6,256,500	16
17	0.9524	333.37	18,700	6,234,000	17
18	0.9459	331.10	18,760	6,211,400	18
19	0.9396	328.89	18,820	6,189,700	19
20	0.9333	326.69	18,880	6,167,900	20
21	0.9272	324.55	18,940	6,147,000	21
22	0.9211	322.42	19,000	6,126,000	22
23	0.9150	320.28	19,060	6,104,500	23
24	0.9091	318.22	19,120	6,084,400	24
25	0.9032	316.15	19,180	6,063,800	25

The heat value may be closely approximated by means of the following formula (Jour. Am. Chem. Soc., Oct., 1908):

$$\text{B.T.U.} = 18{,}650 + 40\,(B - 10),$$

in which B = degrees Baumé.

Oil that is to be transported or stored or used for fuel inside of buildings should be of the "reduced" variety, from which the naphtha and higher illuminating products have been distilled. The gravities of such distillates vary from 20 to 25 degrees Baumé, or close to 0.9 specific gravity, and their flash points range from 240 degrees F. to 270 degrees F. One barrel of crude oil contains 42 gallons and weighs from 310 to 350 pounds, according to the specific gravity. Compared with coal, oil occupies about 50 per cent less space and is 35 per cent less in weight,

for equal heat values. The comparative heat values of coal and oil are approximately as follows:

B.T.U. per Pound of Coal.	Pounds of Coal Equal to 1 Barrel of Oil.	Barrels of Oil Equal to 1 Short Ton of Coal.
10,000	620	3.23
11,000	564	3.55
12,000	517	3.87
13,000	477	4.19
14,000	443	4.52
15,000	413	4.84

49. Efficiency of Boilers with Fuel Oil. — A coal-burning boiler which utilizes 80 per cent of the heat value of the fuel is exceptional — 75 per cent represents very good practice, and 70 per cent a fair average for good practice. The great majority of coal-burning boilers, however, operate at efficiencies less than 65 per cent. With oil fuel a boiler and furnace efficiency of 75 per cent is quite ordinary and 80 per cent not uncommon. This increase in efficiency is partly due to the fact that the oil is readily broken up and brought into immediate contact with the necessary air for combustion and loss due to excessive air dilution is correspondingly reduced.

Table 19 gives the theoretical air requirements for different densities of fuel oils and Table 20 the air excess for various efficiencies. These tables were compiled by C. R. Weymouth (Trans. A.S.M.E., Vol. 30, p. 801).

TABLE 19.

POUNDS OF AIR PER POUND OF OIL AND RATIO OF AIR SUPPLIED TO THAT CHEMICALLY REQUIRED.

Per Cent CO_2 by Volume as Shown by Analysis of Dry Chimney Gases.	Light Oil, C, 84%; H, 13%; S, 0.8%; N, 0.2%; O, 1%; H_2O, 1%.		Medium Oil, C, 85%; H, 12%; S, 0.8%; N, 0.2%; O, 1%; H_2O, 1%.		Heavy Oil, C, 86%; H, 11%; S, 0.8%; N, 0.2%; O, 1%; H_2O, 1%.	
	Lbs. of Air per Lb. of Oil.	Ratio of Air Supply to Chemical Requirements.	Lbs. of Air per Lb. of Oil.	Ratio of Air Supply to Chemical Requirements.	Lbs. of Air per Lb. of Oil.	Ratio of Air Supply to Chemical Requirements.
4	51.40	3.607	51.93	3.704	52.45	3.803
5	41.31	2.899	41.71	2.975	42.12	3.054
6	34.58	2.427	34.90	2.490	35.23	2.554
7	29.77	2.089	30.04	2.143	30.31	2.198
8	26.17	1.836	26.39	1.883	26.62	1.930
9	23.37	1.640	23.56	1.680	23.75	1.722
10	21.12	1.482	21.29	1.518	21.45	1.555
11	19.83	1.391	19.43	1.386	19.58	1.419
12	17.76	1.246	17.88	1.276	18.01	1.306
13	16.46	1.155	16.57	1.182	16.69	1.210
14	15.36	1.078	15.45	1.102	15.55	1.127
15	14.39	1.010	14.48	1.033	14.57	1.056

Table 21 gives the results of a series of tests made at the Redondo plant of the Pacific Light & Power Company, California, on a 604-horsepower B. & W. boiler equipped with Hammel furnaces and burners. The boiler was in regular service and under usual operating conditions.

TABLE 20.

BOILER EFFICIENCY FOR EXCESS AIR SUPPLY (OIL FUEL).

Excess Air Supply, Per Cent.	10	50	75	100	150	200
Assumed temperature of escaping gases, deg. F.	400	450	475	490	Over 500	Over 500
Corresponding ideal efficiency of boiler, per cent	84.2	80.27	77.66	75.22	Under 70.94	Under 67.09
Possible saving in fuel due to reduction of air supply to 10 per cent excess, expressed as per cent of oil actually burned under assumed conditions	0	4.67	7.78	10.68	Over 15.76	Over 20.32

50. Comparative Evaporative Economy of Oil and Coal. — In determining the comparative economy of coal and oil, the fixed and operating charges must be considered in addition to the cost and efficiency of the fuel. From the market quotation on oil and coal and the comparative heating values of each the actual cost per B.t.u. is readily obtained, and by combining this with the relative efficiencies from the furnace standpoint the net cost of the fuel is obtained. The fixed charges vary with the location and size of the plant and are approximately the same per boiler horse power for a given location in both cases. The insurance rates may be greater with the oil fuel and the depreciation of the boiler setting may be somewhat larger, but in a well-constructed furnace the latter item should be the same in both instances for average rates of combustion. The operating charges are decidedly in favor of the oil fuel, since no ash handling is necessary. Oil fuel is readily fed to the furnace, and the cost of attendance may be materially less than with coal firing, and one man may safely control from eight to ten boilers. Table 148, Chapter XVII, gives data relative to the cost of producing electrical power in connection with oil-fired steam plants.

51. Oil Burners. — The function of the burner is to atomize the oil to as nearly a gaseous state as possible.

Classification of a few well-known burners

Mechanical Spray:

Körting.

Vapor or Carburettor:

Durr.

Harvey.

TABLE 21.

EVAPORATIVE TESTS OF OIL-BURNING BOILER.
(Pacific Light & Power Co., Redondo, Cal.)

Date of test (1910)		Aug. 8.	Aug. 9,	Aug. 10.	Aug. 11.	Aug. 12.	Aug. 13.	Sept. 5.	Average of all Tests.
Test number	No.	1	2	3	4	5	6	7	
Duration of test	Hrs.	7	7	7	7	7	7	7	7
Steam pressure	Lbs.	184.9	184.9	186.0	184.7	183.5	184.6	184.9	184.8
Steam temperature	Deg. F.	473.3	457.4	468.8	465.1	473.8	493.8	526.7	479.8
Superheat	Deg. F.	91.8	76.0	86.9	83.7	93.0	112.5	144.3	98.3
Temperature, feed water	Deg. F.	90.6	92.6	92.7	93.4	90.8	94.6	101.2	93.7
Factor of evaporation		1.237	1.225	1.232	1.229	1.237	1.245	1.259	1.238
Average water level	Ins.	4	4	4	4	4	4	4	4
Barometer	Ins.	29.97	30.00	30.00	30.09	30.08	30.04	29.68	29.97
Temperature boiler room	Deg. F.	88.7	86.6	84.4	85.2	87.3	86.7	84.4	86.2
Temperature flue gases	Deg. F.	385.3	397.5	409.1	406.2	429.0	477.1	537.5	434.5
Draft { In ash pit	Ins.	0.025	0.035	0.055	0.044	0.071	0.127	0.230	0.084
Draft { In furnace	Ins.	0.01	0.005	0.025	0.014	0.060	0.130	0.188	0.062
Draft { In rear pass	Ins.	0.005	0.014	0.061	0.046	0.133	0.296	0.471	0.147
Carbon dioxide	Per Cent.	12.2	13.4	13.3	14.3	14.2	13.3	12.1	13.2
Oxygen	Per Cent.	3.6	2.7	2.4	1.8	1.7	2.8	6.8	3.1
Excess air	Per Cent.	28.7	17.7	18.5	10.6	11.3	18.5	43.0	21.2
Smoke		None.	None.	None.	None.	None.	None.	Lt. Haze.	
Position of ash-pit doors		Doors Wide Open Throughout all Tests.							
Temperature in first pass above third tube	Deg. F.	1,100	1,090	1,160	1,180	1,240	1,300	1,600	1,240
Temperature top first pass	Deg. F.	640	640	700	680	780	940	1,170	793
Temperature top second pass	Deg. F.	570	540	620	610	650	740	820	650
Temperature bottom second pass	Deg. F.	500	500	520	510	550	600	700	554
Temperature bottom third pass	Deg. F.	450	450	505	495	530	570	660	523
Total water actually evaporated	Lbs.	85,766	111,988	129,628	129,609	156,622	191,290	226,558	147,351
Total water evaporated from and at 212° F.	Lbs.	106,092	137,185	159,702	159,289	193,741	238,156	285,236	182,771
Water per hour evaporated from and at 212° F.	Lbs.	15,156	19,598	22,814	22,756	27,677	34,022	40,748	26,110
Steam used by burners { Pounds per hour	Lbs.	234	441	549	549	624	708	873	568
Steam used by burners { Per cent of total steam	Per Cent.	1.54	2.25	2.40	2.40	2.25	2.08	2.13	2.15
Steam pressure to burners	Lbs.	48.2	74.7	99.7	102.6	120.4	142.5	167.6	122.2
Oil pressure to burners	Lbs.	11.4	15.2	24.4	25.4	38.3	46.1	61.6	31.6
Temperature of oil to burner line	Deg. F.	131.3	134.3	133.5	142.3	140.1	142.1	141.7	137.9
Kind of oil burned		Oil from Los Angeles Fields.							
Specific gravity of oil at 60° F.	S.G.	0.9770	0.9770	0.9763	0.9770	0.9776	0.9776	0.9797	0.9776
Specific gravity of oil, Beaume, at 60° F.	S.G.	13.3	13.3	13.4	13.3	13.2	13.2	12.9	13.2
Moisture in oil	Per Cent.	0.4	0.5	0.45	0.4	0.8	0.65	0.6	0.54
Heat value of oil as fired	B.t.u.	18,280	18,256	18,131	18,253	18,214	18,171	17,985	18,184
Heat value of oil corrected	B.t.u.	18,353	18,347	18,212	18,326	18,357	18,289	18,093	18,281
Total oil as fired	Lbs.	6,913	8,758	10,322	10,115	12,602	16,580	20,205	12,213
Total oil corrected for moisture	Lbs.	6,885	8,714	10,276	10,075	12,501	16,472	20,084	12,144
Oil per hour as fired	Lbs.	987	1,251	1,474	1,445	1,800	2,369	2,887	1,745
Oil per hour, corrected	Lbs.	983	1,245	1,467	1,439	1,786	2,353	2,869	1,735
Water evaporated F. and A. 212° F. per square F.H.S.	Lbs.	2.58	3.24	3.78	3.77	4.58	5.63	6.74	4.32
Boiler horse power, builders' rating	B.h.p.	604	604	604	604	604	604	604	604
Boiler horse power developed	B.h.p.	439.3	568.0	661.3	659.6	802.2	986.2	1181.0	756.8
Per cent, builders' rating	Per Cent.	72.7	94.0	109.4	109.2	132.8	163.3	195.5	125.3
Water evaporated per pound of oil from and at 212° F. { Oil as fired	Lbs.	15.35	15.66	15.47	15.75	15.37	14.37	14.12	15.15
Water evaporated per pound of oil from and at 212° F. { Corrected for moisture	Lbs.	15.41	15.74	15.54	15.81	15.49	14.46	14.20	15.23
Boiler efficiency	Per Cent.	81.1	82.8	82.4	83.3	81.5	76.4	75.8	80.47

Spray Burners:
- Outside Mixers.
  - (*a*) Peabody.
  - (*b*) Warren.
- Inside Mixers.
  - (*a*) Hammel.
  - (*b*) Kirkwood.
  - (*c*) Branch.
  - (*d*) Williams.

Oil burners for burning liquid fuel may be divided into three general classes:

1. Mechanical spray, in which the oil, previously heated to a temperature of about 150 degrees F., is forced under pressure through nozzles so designed as to break it up into a fine spray. The Körting Liquid Fuel Burner, Fig. 16, is an example of this type. In this design a

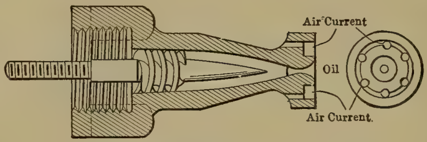


Fig. 16. Körting Fuel-oil Burner.

central spindle, spirally grooved, imparts a rotary motion to the oil and causes it to fly into a spray by centrifugal force on issuing from the nozzle. The particles of oil are burned in the furnace when they come in contact with the necessary air to effect combustion. This type of burner is little used in this country in connection with power-plant work, but is meeting with much success on the continent.

2. Vapor burners, or carburettors, in which the oil is volatilized in a heater or chamber and then admitted to the furnace, are seldom used except in connection with refined oils, as the residuals from crude oil are vaporized only at a high temperature. The Durr and Harvey gasifiers are the best known of this type.

3. Spray burners are by far the most common in use. In this type the oil is held in suspension and forced into the furnace by means of a jet of steam or compressed air. Spray burners are designed either as *outside mixers*, in which the oil and atomizing medium meet outside the apparatus, or *inside mixers*, in which the oil and atomizing medium mingle inside the apparatus.

The *Peabody burner*, Fig. 17, illustrates the principles of the "outside-mixer" type of apparatus. In this type the oil flows through a thin slit and falls upon a jet of steam which atomizes it and forces it into the

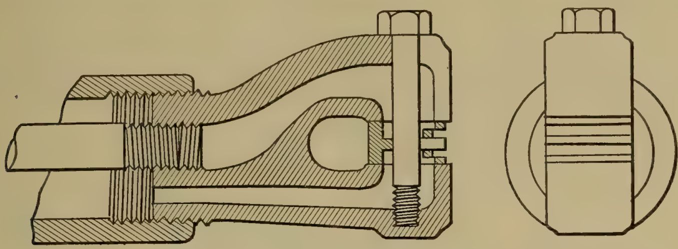


FIG. 17. Peabody Fuel-oil Burner.

furnace in a fan-shaped spray. A feature of this apparatus is its simplicity of construction.

Fig. 18 illustrates the Hammel burner as used at the power house of the Pacific Light and Power Company, Los Angeles, Cal. Oil enters the burner under pressure and flows through opening D to the mouth of the burner, where it is atomized by the steam jets issuing from slots G,

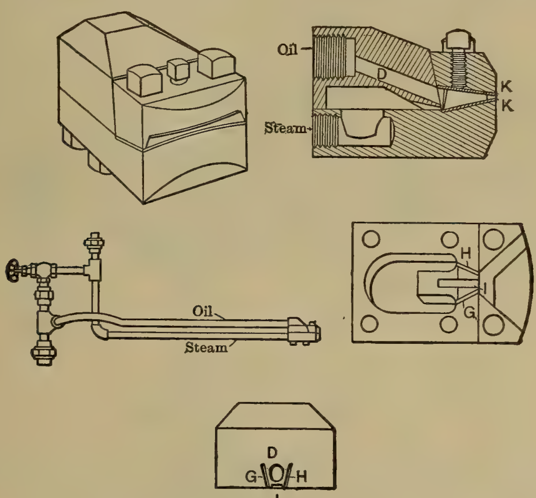


FIG. 18. Hammel Fuel-oil Burner.

H, and I. The oil is preheated to facilitate its flow through the supply system. Plates K-K are removable and are easily replaced when worn out or burned. The Hammel burner belongs to the "inside mixers."

A few well-known types of "inside mixers" are illustrated in Figs. 18 to 20. The operation is practically the same in all of them and they differ only in mechanical details.

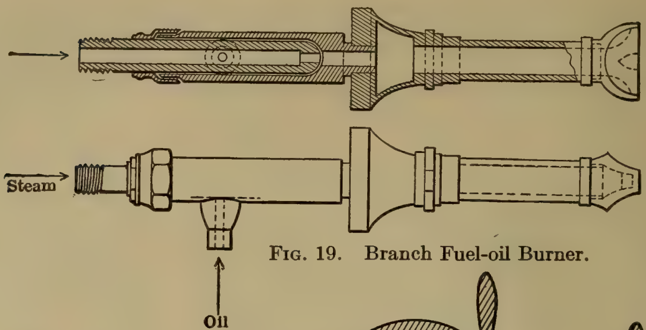


FIG. 19. Branch Fuel-oil Burner.

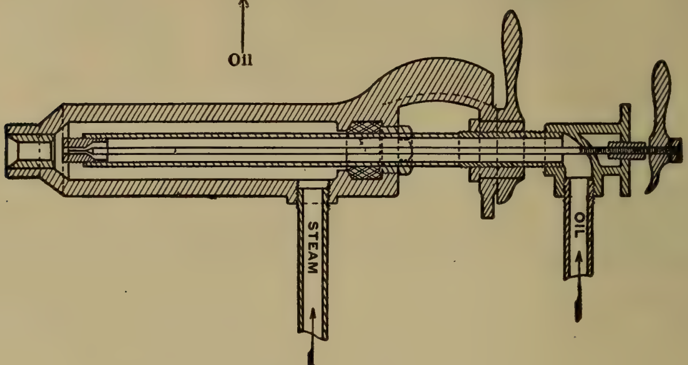


FIG. 20. Kirkwood Fuel-oil Burner.

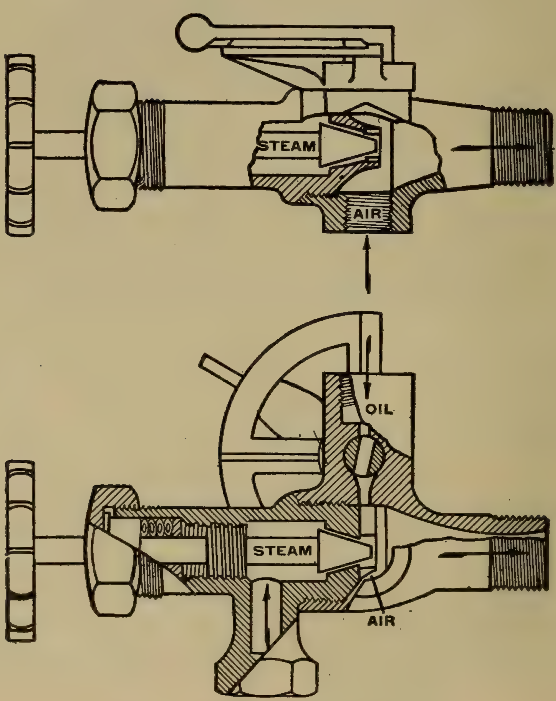


FIG. 21. Williams Fuel-oil Burner.

The Williams burner, Fig. 21, differs somewhat from the others in that the air supply passes through the burner and mingles with the oil and steam before entering the furnace.

The simplest and most reliable burners are of the Hammel type and are much in evidence in the Pacific States.

52. Furnaces for Burning Oil Fuel. — The efficient combustion of oil fuel depends more upon the proportions of the furnace than upon the type of burner, provided, of course, the latter is of modern design. While it is desirable to have incandescent brickwork around the flame it is impossible to do so in many cases and a satisfactory compromise

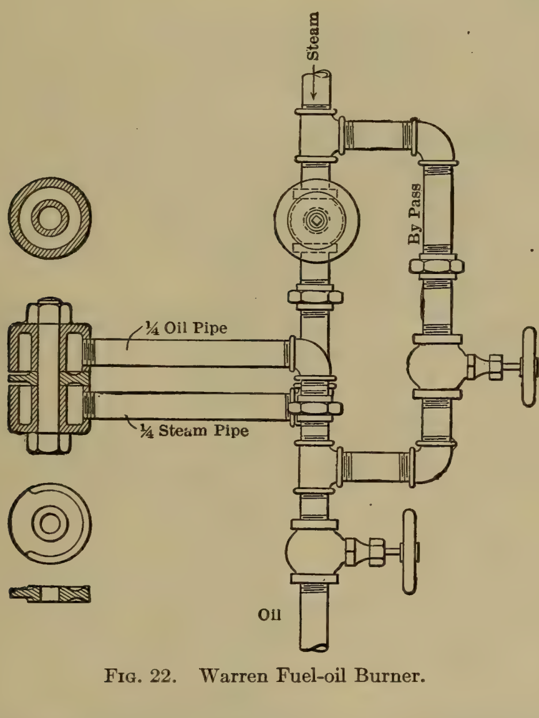


FIG. 22. Warren Fuel-oil Burner.

is effected by using a flat flame burning close to a white-hot floor through which air is steadily flowing. A good burner will maintain a suspended flame clear and smokeless in a cold furnace. The path of the flame in the furnace must be such as to insure uniform distribution of heat over the boiler heat-absorbing surfaces without direct flame impingement. Under ordinary firing the flame should not extend into the tubes. The first pass of the boiler should be located directly over the furnace in order that the heating surface may absorb the radiant energy from the incandescent fire brick. Fire-brick arches and target walls

are not to be recommended on account of the localization of heat resulting in burning out the tubes or bagging the shell and on account of the limited overload capacity.

Fig. 23 shows the general details of a Hammel oil-burning furnace illustrating current practice on the Pacific coast. The burner tip is housed in a slot located in the back of an arched recess in the bridge

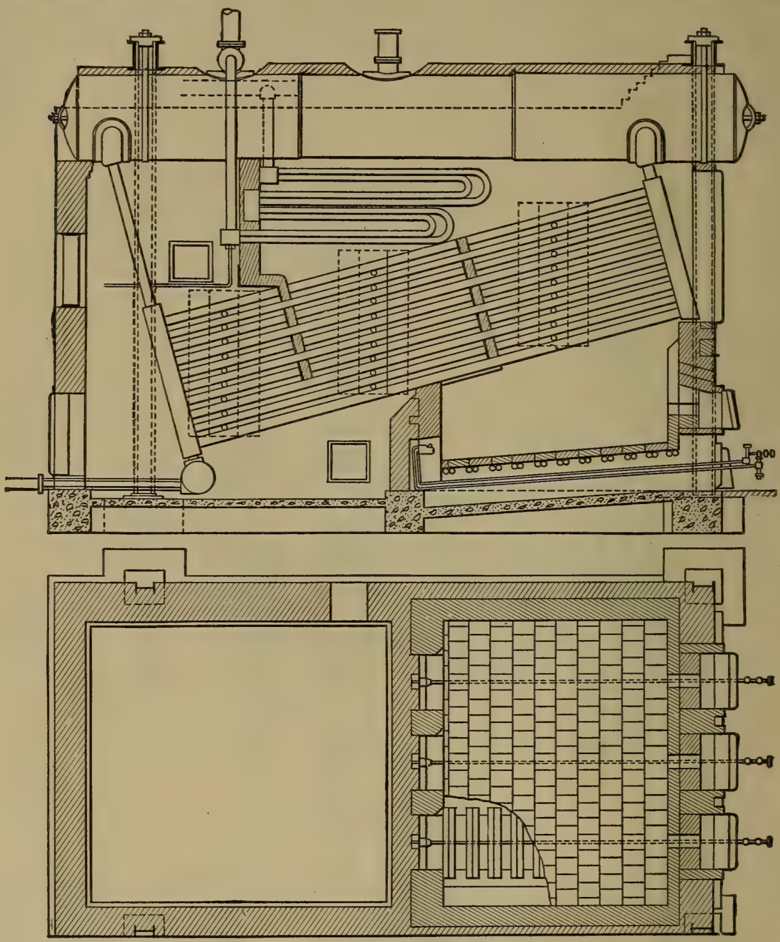


FIG. 23. Furnace for Burning Fuel Oil, Rear Feed (Hammel).

wall and the flame is projected forward toward the front of the furnace. The furnace floor is carried on pieces of old two-inch pipe or on old rails and is solid except for narrow air slots through the deck and in front of each arch. Each burner with its accompanying recess has a separate air tunnel from the boiler front; these tunnels do not commu-

nicate with each other under the furnace floor and by closing the ash-pit door any tunnel can be sealed up while the others are supplying air to their particular burners. The Hammel furnace is a modification of

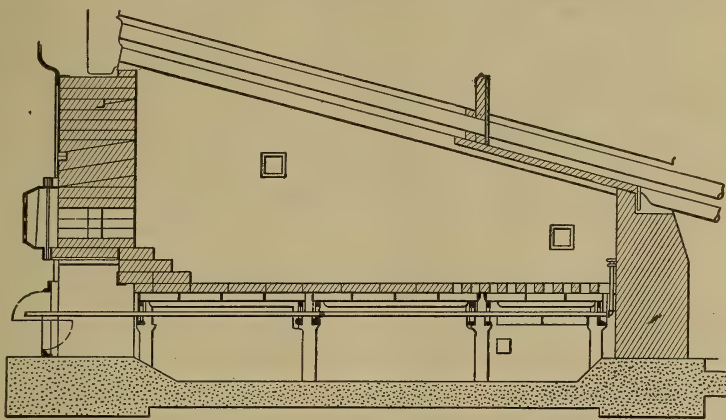


Fig. 24.—Peabody Fuel Oil Furnace.

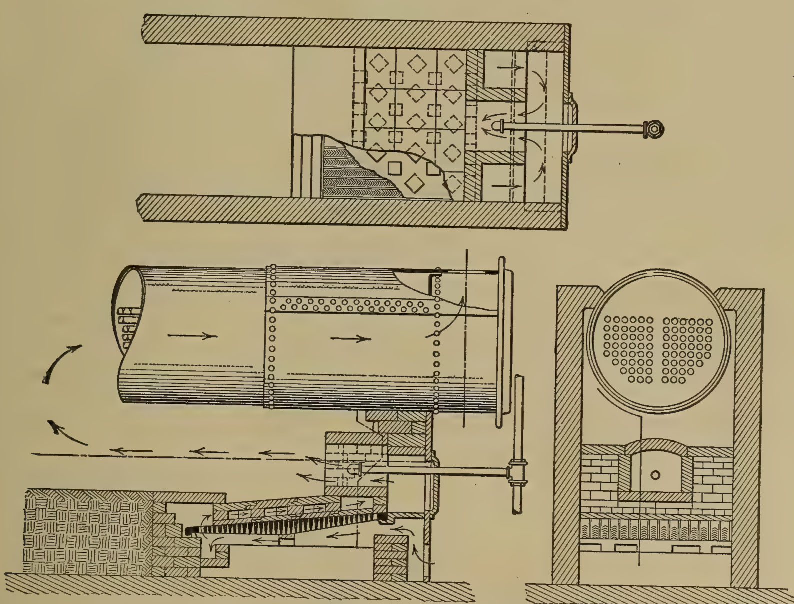


Fig. 25.—Modern Furnace for Burning Fuel Oil, Front Feed.

the well-known Peabody furnace, a section through which is shown in Fig. 24.

Fig. 25 gives the general details of a modern oil-burning furnace, with front feed, as applied to a horizontal return tubular boiler.

TABLE 22.

TESTS OF FUEL-OIL BURNERS.

Number of Test.	Authority of Test.	Reference.	Type of Burner.	Steam Pressure (Pounds per Square Inch Gauge).	Pressure of Medium used for Spraying Oil.	Temperature of Spraying Medium (Degrees F.).	Equivalent Evaporation from and at 212° F. (Pounds).	Steam used in Spraying Oil; Per Cent of Total Steam Generated.	Efficiency of Boiler (Per Cent).
1	United States Naval Board.		O. C. B. W. (Air)	273.5	3.2	......	12.7	2.38	62.8
2		Report of U. S. Naval Liquid Fuel Board. 1902.	do	273.5	4.62	......	12.18	3.89	60.3
3			do	273.5	0.78	102.5	14.43	1.06	71.5
4			do	273.5	3.37	122	11.73	2.88	58.1
5		Engineer U. S., June 1, 1903	do	273.5	1.41	120	14.22	1.97	70.4
6			do	271.5	1.31	113.5	14.12	1.53	69.9
7		Engineering Record, December 20, 1902.	do	272.5	4.66	161	13.29	7.45	65.8
8			do	276	4.68	136	10.77	4.25	53.4
9		Western Electrician, December 13, 1902.	Hayes (Steam)	273.5	32	......	13.89	5.77	68.9
10			O. C. B. W. (Steam)	273.1	29.9	444	13.47	3.98	66.7
11		Engineer, London, Nov. 14, 1902.	do	273.7	61.4	408	13.45	4.41	66.7
12			do	274.2	81	401	13.58	5.03	67.3
13			Reed (Air and Steam)	276.7	92	375	14.35	8.54	71.1
14			do	277.4	89	416	14.06	6.09	69.7
15	Wallsend	Engng., Nov. 6, 1903	Körting*	113	75	240‡	14.45	......	......
16	Denton	Power, February, 1902	Williams (Steam)	86.5	......	......	15.49	3.5	78.5
17	Pacific Light and Power Co., Los Angeles, Cal.	Tests by B. & W. and Stirling Companies.	H. & M. (Steam)	156	36†	109‡	15.66	2.33	80.6
18			Hammel (Steam)	156	35†	88‡	14.87	2.72	70.7
19			do	156.4	36†	109‡	15.85	1.96	83
20	B. R. T. Collins	Jour. A.S.M.E., Aug., 1911	Peabody (Steam)	142	......	97	14.38	1.54	81

* Uses no steam or air. Oil under pressure. † Oil pressure. ‡ Temperature of oil.

53. Atomization of Oil. — For efficient combustion the oil should be injected into the furnace in the form of a spray. Three systems of atomization are in use in stationary practice, namely, mechanical, air, and steam. Of these, by far the greater number of installations in the United States are of the last order.

The mechanical or Körting system is not much in evidence in this country, but it is used extensively in Europe. The operation of this system is described in paragraph 51. The makers state that to operate the pumps and supply the heat to the oil takes from $\frac{3}{4}$ to 1 per cent of the steam evaporated. Mechanical atomization presents many possibilities and it is not unlikely that future development may lie along this path.

In air atomization the air is used at pressures from $1\frac{1}{2}$ to 60 pounds per square inch, depending upon the type of burner. From Table 22 it will be seen that the total steam used to compress the air varies from 1.06 to 7.45 per cent of the total generated. For air atomization and with air pressures of from 20 to 30 pounds per square inch, J. H. Hoppes, (Jour. A.S.M.E., Aug., 1911, p. 902), states that from 6 to 10 cubic feet of air per minute per pound of oil burned will be required. Compressed air offers no opportunity for fuel saving over the use of steam direct in cases where steam is available. In certain industrial operations where high temperatures are essential the use of air is preferred. When it is necessary to use high-pressure air the economy decreases with the increase in pressure, since the cost of each cubic foot of compressed air increases rapidly with the pressure, but its ability to atomize the oil does not increase proportionately.

Steam is the most commonly used medium for atomizing the oil, since its use obviates complication and risk of interrupted service. The amount of steam required to atomize the oil varies from 0.3 to 0.7 pounds per pound of oil, with an average of about 0.5 pounds. The steam consumption is generally stated in per cent of the total steam generated, but the results are misleading since the percentage factor depends largely upon the efficiency of the boiler. Table 22 gives the results of a number of tests of different types of burners with air and steam as an atomizing medium.

54. Oil-feeding Systems. — Fig. 26 gives a diagrammatic arrangement of the piping commonly employed in feeding oil fuel to the burners. Steam-actuated oil pumps, installed in duplicate, draw the fuel from the supply tank and deliver it under pressure to the burners. The piping is cross-connected so that repairs can be made without interrupting the service. The oil is heated from the pump exhaust before it is supplied to the burners. This should not be carried beyond the

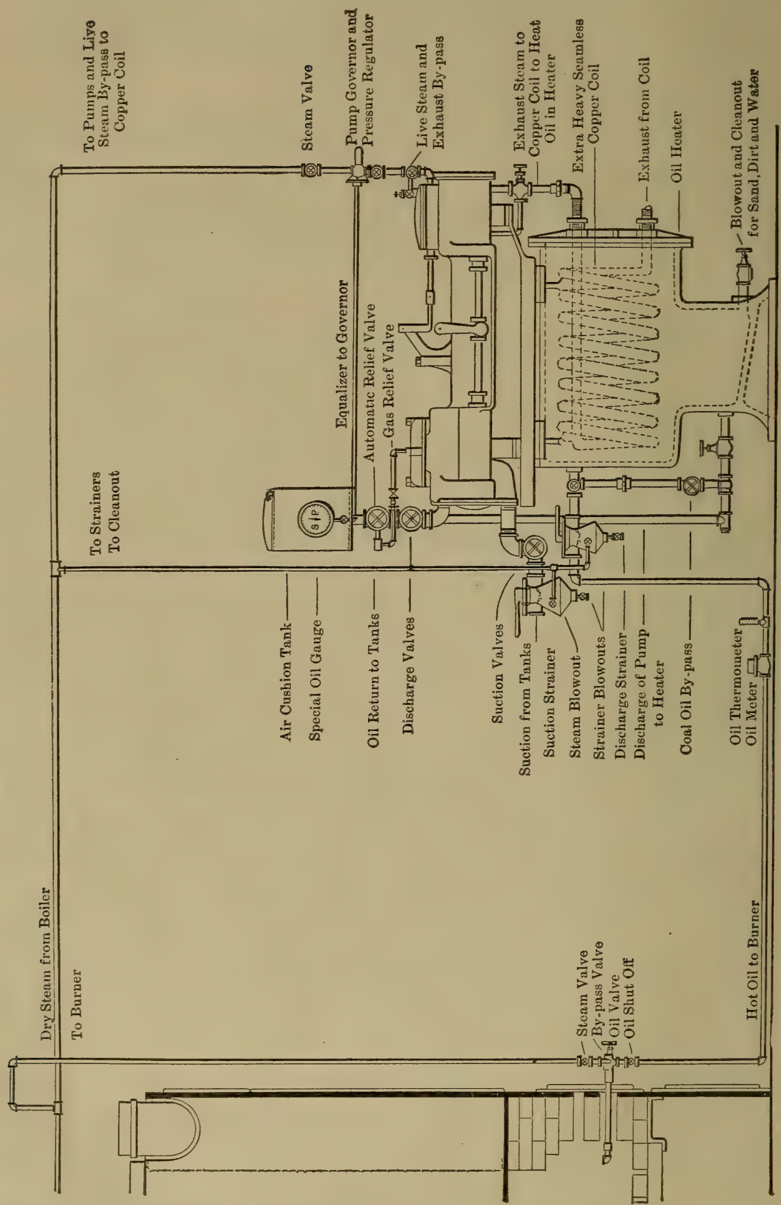


FIG. 26. —Diagrammatic Arrangement of Piping for Oil Fuel System.

flash point of the oil used or there will be danger from carbon deposits in the supply pipe. A strainer is placed in the suction line between the storage tank and the oil-pressure pump to minimize clogging of the burner. In some instances strainers are also placed in the supply pipe between the heater and burner. The relief valve between the pumps and burners is set at a definite maximum oil pressure so as to prevent excessive pressure. The oil meter is for the purpose of checking the storage tank indicator. All oil piping is installed so that it can be drained back to the storage tank by gravity in case of necessity. In many large plants the strainers, meters, heaters and piping are installed in duplicate. Arrangements are usually made for the oil to be delivered at constant pressure. The supply of steam to the burner is controlled by regulating the pressure in a separate main common to all burners, the pressure in the main bearing a certain predetermined relation to the pressure in the oil mains. In most installations the supply of steam and oil at the burner is regulated by hand to meet the requirements of the individual burners. At the Redondo plant of the Pacific Light and Power Company, Redondo, Cal., the supply of oil and steam to all burners and the supply of air for combustion to any number of boilers are automatically controlled from a central point. For a description of this system see Trans. A.S.M.E., Vol. 30, p. 808.

Low-pressure systems are ordinarily operated under standpipe pressures as in Fig. 27, which illustrates the arrangement of apparatus as advocated by the International Gas and Fuel Company. A steam pump B draws the oil from the buried tank through pipe Z and delivers it to the standpipe E. Thence it flows through pipe I to the burners under a head of about 10 feet. The pump runs constantly, the surplus oil flowing back to the tank through the pipe T. The oil is heated by the exhaust pipe Z'. The oil pump is provided with a device D having a piston connected by a chain with a cock S, which automatically opens when the boiler is not under steam pressure, so that the standpipe will be emptied, the oil flowing to the storage tank.

Fig. 28 illustrates the Hydraulic Oil Storage Company's system of storing oil and delivering it to the burners. The oil reservoirs are placed below grade, as indicated, to minimize fire risk. The operation is as follows: Water enters the "float box" and flows through a "three-way cock" to the bottom of the reservoir until all of the oil and water-pipes are filled up to the level of the float box, when the float automatically cuts off the supply. This flooding of the entire system drives out all of the air. The three-way cock is then turned to "discharge" and part of the water flows to the sewer. The tank car or wagon is

next attached to the "oil inlet" and the oil flows into the tank and displaces the water until the level of the "filler float" is reached, when the supply is automatically cut off. The inlet is so placed that the head of oil in the tank car is sufficiently great to overcome the opposing head of water. The three-way valve is next turned to the first position and the head of water forces the oil to the burners. After the oil has been withdrawn from the storage tank the water can only rise to the level of the water in the float box and therefore cannot be fed to the furnace. The small steam pipe admits steam into the tank and heats the oil, thereby making it flow more freely.

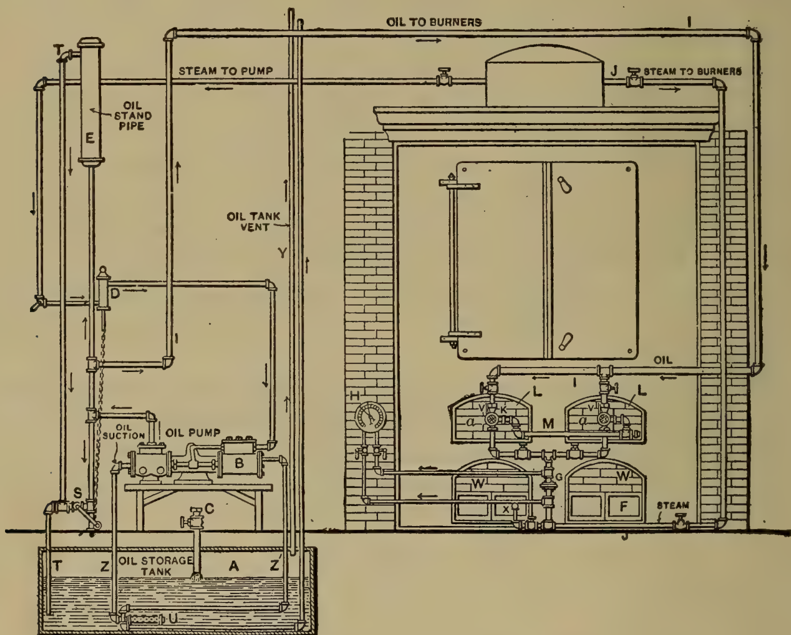


FIG. 27. International Gas and Fuel Company's Fuel Oil System.

55. Oil Transportation and Storage. — Fuel oil is delivered in bulk, either in tank cars, barges or steamships, or by pipe lines, depending upon the location of the plant. It must be stored in accordance with underwriters' requirements and community ordinances. In outlying districts the storage reservoirs may be placed above the ground, but in cities they must be located underground. In small plants surface tanks are sometimes constructed of wood but in the majority of installations both surface and underground tanks are constructed of steel plate, plain concrete or reinforced concrete. In large plants the greater portion of the oil is stored in surface tanks some distance away

from the plant. From these main reservoirs the oil is pumped into auxiliary tanks or vaults outside the building, but beneath the boiler-room level. In no case is the oil permitted to gravitate to the burners but must reach them by means of pumps. Storage tanks should be fitted with vent pipes; return and overflow pipes; indicators showing the depth of oil; steam coils for preliminary heating in cold weather, or, for thick, viscous oils, steam pipes for smothering the flames in case of fire, and suitable manholes for cleaning out purposes. To conform with underwriters' requirements the tops of the tanks should be placed

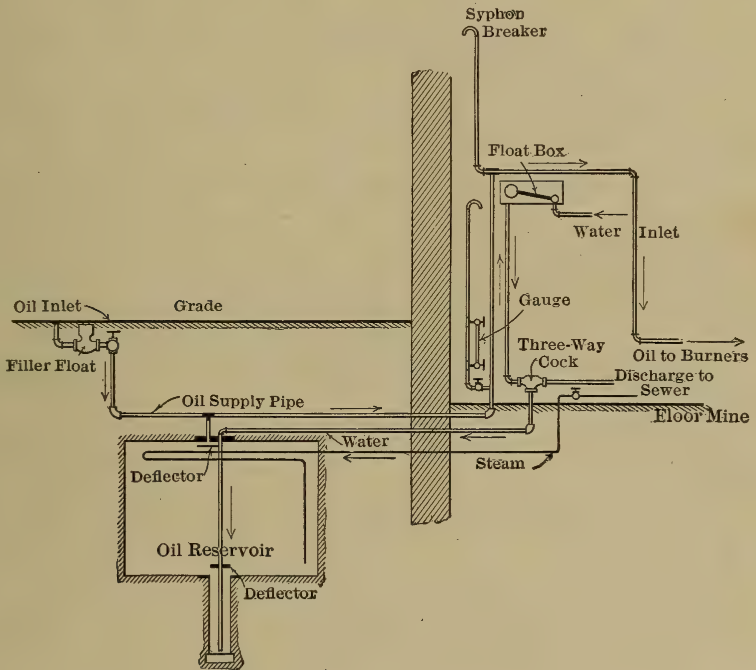


FIG. 28. Hydraulic Oil Storage Company's Fuel Oil System.

below the level of the lowest pipe used in connection with the apparatus. Steel tanks, including foundations, roof and setting, cost approximately $1.00 per barrel, while concrete vaults and forms cost approximately fifty cents per cubic foot of concrete exclusive of excavation.

56. The Purchase of Fuel Oil. — The following extracts from Bulletin No. 3, 1911, Bureau of Mines ("Specifications for the Purchase of Fuel Oil for the Government, with Directions for Sampling Oil and Natural Gas"), though primarily intended for the guidance of Government officials, may be of service to engineers:

1. In determining the award of a contract, consideration will be given to the quality of the fuel offered by the bidders, as well as the price, and should it appear to be the best interest of the Government to award a contract at a higher price than that named in the lowest bid or bids received, the contract will be so awarded.

2. Fuel oil should be either a natural homogeneous oil or a homogeneous residue from a natural oil; if the latter, all constituents having a low flash point should have been removed by distillation; it should not be composed of a light oil and a heavy residue mixed in such proportions as to give the density desired.

3. It should not have been distilled at a temperature high enough to burn it nor at a temperature so high that flecks of carbonaceous matter began to separate.

4. It should not flash below 60 degrees C. (140 degrees F.) in a closed Abel-Pensky or Pensky-Martens tester.

5. Its specific gravity should range from 0.85 to 0.96 at 15 degrees C. (59 degrees F.); the oil should be rejected if its specific gravity is above 0.97 at that temperature.

6. It should be mobile, free from solid or semi-solid bodies, and should flow readily at ordinary atmospheric temperatures and under a head of 1 foot of oil, through a 4-inch pipe 10 feet in length.

7. It should not congeal nor become too sluggish to flow at 0 degree C. (32 degrees F.).

8. It should have a calorific value of not less than 10,000 calories per gram (18,000 B.t.u. per pound); 10,250 calories to be the standard. A bonus is to be paid or a penalty deducted according to the method stated under section 21, as the fuel oil delivered is above or below this standard.

9. It should be rejected if it contains more than 2 per cent water.

10. It should be rejected if it contains more than 1 per cent sulphur.

11. It should not contain more than a trace of sand, clay or dirt.

12. Each bidder must submit an accurate statement regarding the fuel oil he proposes to furnish. This statement should show:

(*a*) The commercial name of the oil.

(*b*) The name or designation of the field from which the oil is obtained.

(*c*) Whether the oil is a crude oil, a refinery residue, or a distillate.

(*d*) The name and location of the refinery, if the oil has been refined at all.

For sampling, analysis, etc., consult complete bulletin.

Analyses of California Petroleums: Bulletin No. 19, U. S. Bureau of Mines, 1912.

Atomization: Jour. A.S.M.E., Aug. 11, 1911, p. 883, 902; Jl. El. Power and Gas, Dec. 23, 1911.

Burners: Jl. El. Power and Gas, Dec. 23, 1911, Apr. 1, 1911; Engng., Feb. 16, 1912.

Comparative Evaporative Value of Coal and Oil: Jl. El. Power and Gas, March 18, 1911; Jour. A.S.M.E., Aug. 11, 1911, p. 872.

Draft Requirements for Burning Oil Fuel: Jour. A.S.M.E., Aug., 1911; Oct., 1912.

Economy Tests with Oil Fuel: Trans. A.S.M.E., 30–1908, p. 775; Jl. A.S.M.E., Aug. 11, 1911, p. 940.

Furnaces for Burning Oil Fuel: Jl. El. Power and Gas, Dec. 30, 1911, Apr. 8, 1911; Jour. A.S.M.E., Aug., 1911, p. 879; Ir. Td. Review, June 3, 1908; Power, June 16, 1908.

Oil for Steam Boilers: Jour. A.S.M.E., Aug., 1911, p. 931; Jl. El. Power and Gas, Dec. 16, 1911; Power, Aug., 1908, p. 943; Jan. 23, 1908, p. 980; Bulletin No. 131, Louisiana State University.

Precautions with Oil Fuel: Eng. and Min. Jour., Apr. 1, 1911, p. 653.

Purchase of Fuel Oil for the Government: Bulletin No. 3, Bureau of Mines, 1911.

Regulation of Oil Supply to Burners: Trans. A.S.M.E., 30–1908, p. 804.

Storage and Transportation: Jl. El. Power and Gas, Dec. 16, 1911, p. 564; Eng. News, Sept. 25, 1902, p. 232; Power, July 16, 1908.

Unnecessary Losses in Firing Fuel Oil: Trans. A.S.M.E., 30–1908, p. 797.

57. Gaseous Fuels. — These fuels offer all of the advantages of liquid fuels and but few of the disadvantages. The gases most commonly met with in connection with steam power plants are outlined in Table 23. The artificial gases for steam purposes are prohibitive in cost in most cases, and even in blast-furnace installations, where the gases are waste products, the gas engine has virtually supplanted the steam engine for power purposes. In the immediate locality of natural-gas wells gas-fired furnaces may prove to be more economical than coal furnaces, but the limited supply limits its use as a general fuel. From the market quotations on coal and gas and the comparative heating value of each the actual cost per B.t.u. is readily obtained, and by combining this with the relative efficiencies from the furnace standpoint the net cost of the fuel is obtained. The following table, based upon the assumption that one cubic foot of natural gas under standard conditions has a heating value of 1000 B.t.u., will enable an approximate comparison to be made:

B.T.U. per Pound of Coal.	Pounds of Coal Equal to 1,000 Cu. Ft. of Gas.	No. of 1,000 Cu. Ft. of Gas Equal to One Short Ton of Coal.
10,000	100	20
11,000	91	22
12,000	83	24
13,000	77	26
14,000	71	28
15,000	67	30

In burning natural gas under a boiler the furnace requirements are practically the same as for liquid fuel. The burners, of course, will differ in design, since atomization is unnecessary. The majority of patented gas burners are operated on the same principle as the common gas-stove burner. A crude form often used consists of a $\frac{1}{2}$-inch gas pipe placed within a $2\frac{1}{2}$-inch pipe which is bricked in the fire-door opening. A properly installed gas-fired furnace should be capable of converting 72 per cent of the heat value of the fuel into steam, corresponding to approximately 80 per cent of the lower heating value.

Fig. 29 shows a section through a small experimental boiler designed by Prof. Wm. A. Bone, University of Leeds, England, which involves the principle of so-called "surface combustion," and for which

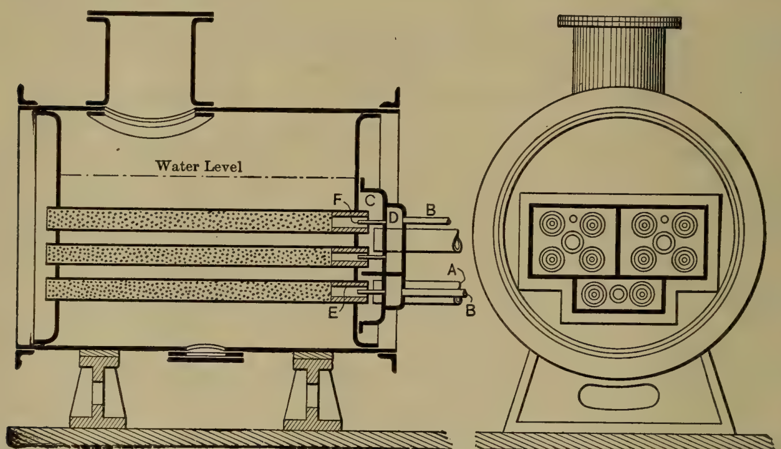


Fig. 29. Experimental Boiler Involving the Principles of "Surface Combustion."

extravagant claims have been made as regards efficiency and capacity. It consists essentially of a plain tubular boiler, having ten tubes, 3 inches in internal diameter. Each of these is bushed with a short tube, *E*, of fire clay and is filled for the rest of its length with finely broken refractory material. Mixing chambers of special design are attached to the front plate of the boiler as indicated. The mixture fed into the boiler tubes from these mixing chambers consists of the combustible gas with a proportion of air very slightly in excess of that theoretically required for combustion. The mixture is injected or drawn in through the orifice in the fire-clay plug. The gas burns without flame in the front end of the tube, the incandescent mass being in direct contact with the heating surface. The combustion of the mixture in contact with the incandescent material is completed before it has traversed a length of 6 inches from the point of entry of the tube.

TABLE 23.

CHARACTERISTICS OF GASEOUS FUELS.

(Ulbright and Torrance; Power, Aug. 6, 1912).

Gas.	Average Constituents of Gas — in Per Cent Volume.									Theoretical Calorific Value per Cu. Ft. of Gas at 62° and Atmospheric Pressure, B.t.u.		Calorific Value per Cu. Ft. by test, B.t.u.		Theoretical Air per Cu. Ft. Gas.	B.t.u. per Cu. Ft. of Mixture.		Calculated B.t.u. per Cu. Ft. of Mixture, Theoretical Air.		Calculated B.t.u. per Cu. Ft. of Mixture — Excess Coef. as Below.				
	CO_2	CO	H	CH_4	C_2H_6	C_2H_4	C_6H_6	N	O	High.	Low.	High.	Low.		High.	Low.	High	Low.	1.1	1.2	1.3	1.4	1.5
Producer:																							
Anthracite	6.03	22.38	13.38	1.96	.....	0.36	.....	55.65	0.803	138.4	125.7	142	131.3	1.053	67.9	64.1	67.5	61.3	58.2	55.4	53	50.8	48.7
Bituminous	9.12	17.54	11.73	4.28	.....	0.36	.....	57.24	0.265	137	125	154	136	1.120	(72.7)	47.8	64.6	59.0	56	53.4	50.8	48.7	46.6
Coke	4.90	27.30	10.07	1.18	.....	.....	.....	56.60	0.550	128.7	126.3	133.7	(125)	0.985	(67.4)	(63)	65.0	63.7	60.7	57.8	55.4	53.1	51.0
Lignite	9.43	18.90	15.13	3.65	.....	0.424	.....	52.50	0.582	144	134	156.4	.....	1.145	(72.8)	......	67.2	62.5	59.3	56.4	53.8	51.5	49.3
Oil	4.10	11.40	5.57	5.87	.....	3.10	.....	66.90	3.060	176	151	......	.....	0.990	....	......	88.4	75.9	72.2	69.0	66.0	63.2	60.7
Peat	12.40	21.00	18.50	2.20	.....	0.40	.....	45.50	0.000	153	141	175	.....	1.170	(80.6)	......	70.6	65.0	61.6	58.8	56.3	53.5	51.2
Wood	13.90	20.03	21.00	2.79	.....	0.60	.....	41.80	0.185	137	128.7	138.7	.....	1.070	(67.0)	......	66.2	62.2	59.2	56.3	53.8	51.5	49.4
Illuminating:																							
Water	4.72	34.8	48.81	4.06	.....	1.97	.....	7.16	0.503	322.1	304	329	278	2.63	(90.6)	(76.6)	88.8	83.8	78.0	73.2	68.8	64.8	61.5
Carbureted water	2.07	24.1	32.40	23.40	.....	12.52	.....	3.75	0.510	646.4	589	677	632	6.84	99.1	90.94	82.4	75.1	69.0	63.9	59.5	55.7	52.3
Coal	1.21	6.18	43.94	37.78	.....	5.87	4.16	3.50	0.502	677	608	655	592	6.00	97.8	86.40	96.7	86.8	80.0	74.2	69.2	64.7	60.8
Natural:																							
Average	0.684	0.647	10.89	79.67	.....	1.26	6.40	5.89	0.79	948	853	965	855	9.85	96.2	87.2	95.2	85.7	78.5	72.7	67.5	63.0	59.1
High hydrogen	0.580	0.730	20.56	50.30	.....	2.07	5.88	8.80	1.13	923	834	895	810	8.66	(92.6)	(83.8)	95.5	86.3	79.2	73.1	68.0	63.5	59.5
Low hydrogen	0.800	0.525	1.92	91.40	.....	0.515	.....	3.25	0.50	967	870	971	862	9.20	96.2	87.2	94.7	85.2	78.3	72.4	67.1	62.7	58.8
Blast furnace	11.80	26.75	3.40	0.30	.....	.....	.....	58.8	0.182	98.7	95.2	98.7	.....	0.735	55.3	54.8	56.8	54.8	52.7	50.5	48.6	46.8	45.2
Coke oven	2.1	6.51	51.56	32.9	(1.4)	2.5	.....	4.0	0.36	577.0	487.0	598.0	506	4.76	96.7	84.0	100.0	84.5	78.2	72.4	67.8	63.5	59.8
Oil gas	2.79	4.96	17.70	23.60	18.78	5.53	.....	37.55	0.702	595.0	542.0	662.0	.....	4.94	(111.3)	......	100.0	91.2	84.2	78.2	73	68.4	64.4

Although the core of the material at this part of the tube is incandescent the heat transference is so rapid that the walls of the tubes are considerably below red heat. The evaporation in regular working order is over 20 pounds per square foot of heating surface and this can be increased 50 per cent with a reduction in efficiency of only 5 or 6 per cent. The figures given by Prof. Bone for the boiler and economizer are as follows:

Date, Dec. 8, 1910.	
Pressure of mixture entering boiler tubes, inches of water	17.3
Pressure of products entering economizer, inches of water	2.0
Steam pressure, pounds per square inch gauge	100.0
Temperature of steam in boiler, degrees F.	334.0
Temperature of gases leaving boiler, degrees F.	446.0
Temperature of gases leaving economizer, degrees F.	203.0
Temperature of water entering economizer, degrees F.	41.9
Temperature of water leaving economizer, degrees F.	136.4
Evaporation per square foot of heating surface per hour, pounds	21.6
Gas consumption, cubic feet per hour, at 32 degrees F. and 14.7 pounds per square inch	996.0
B.t.u. per standard cubic foot (lower heat value)	562.0
Water evaporated per hour from and at 212 degrees F., pounds	550.0
Efficiency of boiler and economizer (on basis of low heat value), per cent	94.3

For further details of Prof. Bone's experiment see American Gas Light Journal, Dec. 4, 1911; Engineering, April 14, 1911; Engineer (London), April 14, 1911. See also editorial, Industrial Engineering, Jan., 1912, p. 59.

CHAPTER III.

BOILERS.

58. As affecting fuel economy the boiler equipment is by far the most important part of the power plant and involves the largest share of the operating expenses. It matters little how elaborate, modern, or well designed it may be, skill, good judgment, and continued vigilance are required on the part of the operator to secure the best efficiency.

Of the various types and grades of boilers on the market experience shows that most of them are capable of practically the same evaporation per pound of coal, provided they are designed with the same proportions of heating and grate surface and are operated under similar conditions. They differ, however, with respect to space occupied, weight, capacity, first cost, and adaptability to particular conditions of operation and location.

59. Classification. — As to design and construction there is an almost endless variety of boilers and furnaces, classified as *internally* and *externally fired; water tube* and *fire tube; through tube* and *return tubular; horizontal* and *vertical.*

The internally fired type includes the *vertical tubular, locomotive, Scotch-marine,* and practically all *flue boilers.* The externally fired includes the *plain cylinder,* the *through tubular, return tubular,* and nearly all stationary *water-tube* boilers.

60. Vertical Tubular Boilers. — Vertical tubular boilers, Figs. 1 and 30, are commonly used where small power, compactness, low first cost, and sometimes portability are the chief requirements, though they are not necessarily restricted to small sizes. The tubes are sometimes arranged so that the spaces between them radiate from a hand hole on one side so that a scraper may readily be inserted to clean the top of the furnace plate. The hand hole in the water leg permits removal of the scale. It is convenient to place a chain in the bottom of the water leg, which can be worked around through the hand hole for the purpose of loosening up the scale deposit. The distance between the furnace crown and top of the grate is never less than 24 inches even in the smallest boiler and should be as great as possible to insure good combustion. Two styles of vertical boilers are in common use, the ordinary vertical type, Fig. 1, and the submerged type, Fig. 30. In

the former the upper tube sheet and part of the tubes are above the water line, and while this feature may tend to superheat the steam to a slight extent, the difficulty from unequal expansion and liability to overheating is of sufficient moment to justify the use of the submerged type, particularly where the boiler is likely to be forced above its rated capacity. The advantages of this type of boiler are: (1) compactness and portability; (2) requires no setting beyond a light foundation; (3) is a rapid steamer, and (4) is low in first cost. The disadvantages are: (1) inaccessibility for thorough inspection and cleaning; (2) small steam space, which results in excessive priming at heavy loads; (3) poor economy except at light loads, as the products of combustion escape at a high temperature on account of the shortness of the tubes; (4) smokeless combustion practically impossible with bituminous coals; (5) the small water capacity results in rapidly fluctuating steam pressures with varying demands for steam.

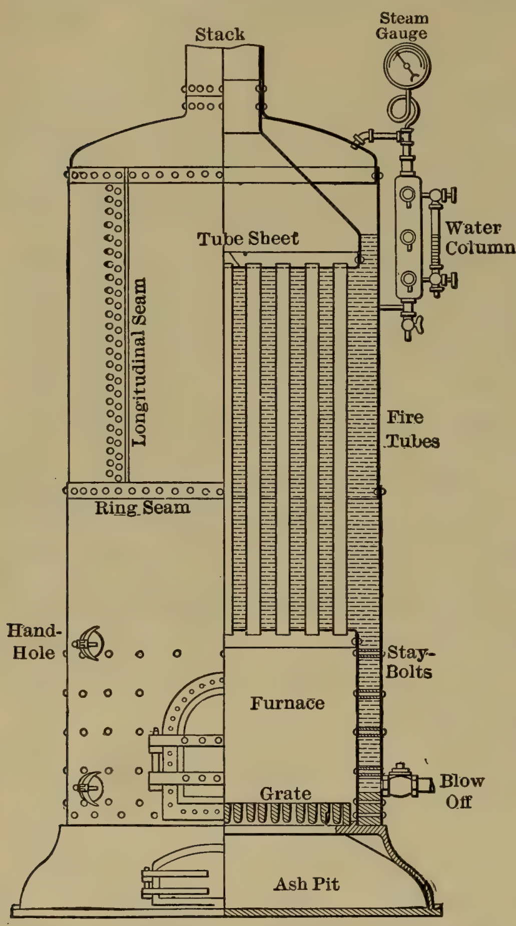
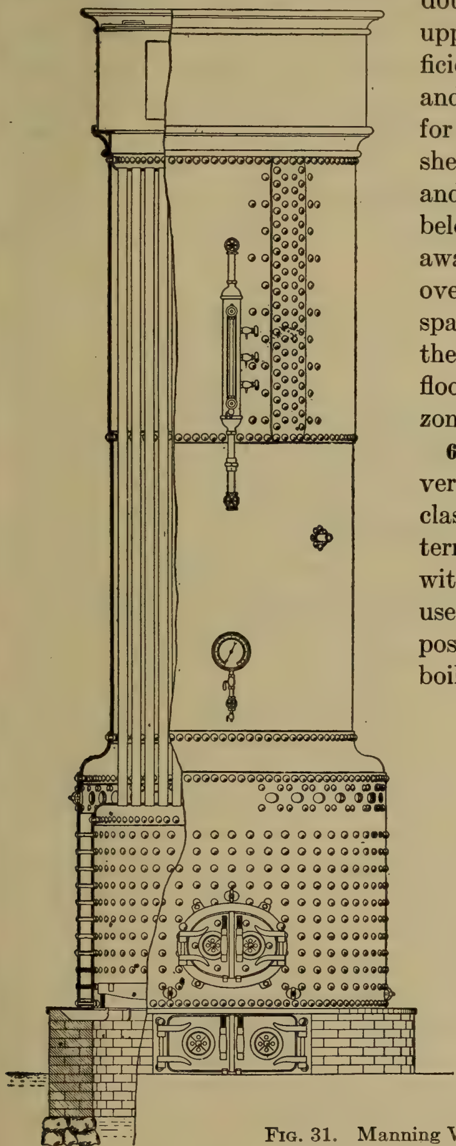


FIG. 30. Vertical Tubular Boiler with Submerged Tube Sheet.

Although vertical fire-tube boilers are usually of very small size, being seldom constructed in sizes over 60 horse power, an exception is found in the Manning boiler, Fig. 31, which is constructed in sizes as large as 250 horse power. Many of the disadvantages found in the smaller types are obviated in the Manning boilers, which, as far as safety and efficiency are concerned, rank with any of the other first-class types. They differ from the boiler described above mainly in having the lower or furnace portion of much greater diameter than the

upper part which encircles the tubes. This permits a proper proportion of grate, which is not obtainable in boilers like Figs. 1 and 30. The double-flanged head connecting the upper and lower shells allows sufficient flexibility between the top and bottom tube sheets to provide for unequal expansion of tubes and shell. The ash pit is built of brick and the water leg does not extend below the grate level, thus doing away with dead-water space. Where overhead room permits and ground space is expensive, this boiler offers the advantage of taking up a small floor space as compared with horizontal types.

61. Fire-box Boilers. — Although vertical fire-tube boilers may be classed as fire-box boilers, yet the term "fire box" is usually associated with the locomotive types, whether used for traction or stationary purposes. The usual form of fire-box boiler as applied to stationary work



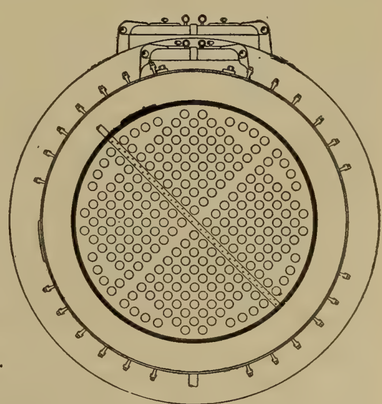


FIG. 31. Manning Vertical Fire-tube Boiler.

is illustrated in Fig. 32. The shell is prolonged beyond the front tube sheet to form a smoke box. The front ends of the tubes lead into the smoke box and the rear ends into the furnace or fire box. The fire box

is ordinarily of rectangular cross section, and is secured against collapse by stay bolts and other forms of stays. In Fig. 32 the smoke box is of cylindrical cross section and hence requires no staying except at the flat surface. Fire-box boilers are used a great deal in small heating plants where space limitation precludes other types. Their steam capacity gives them an advantage over the vertical tubular form. Being internally fired no brick setting is required. They are usually

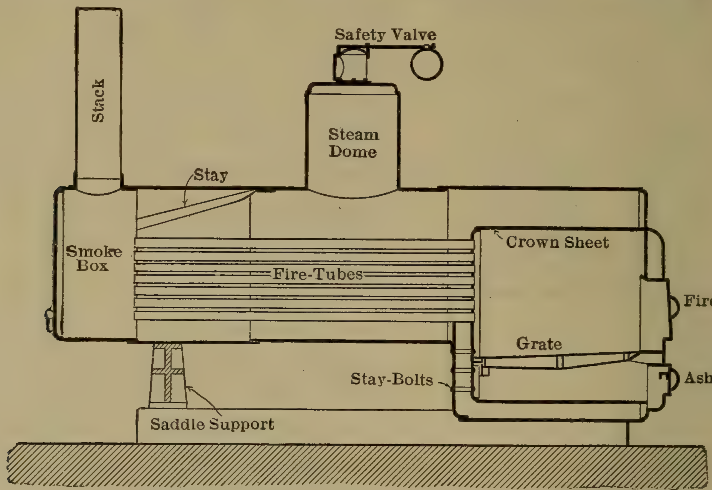


Fig. 32. Typical Fire-box Boiler — Stationary Type.

of cheap construction, designed for low pressure, and seldom made in sizes over 75 horse power. Unless carefully designed and constructed high steam pressures are apt to cause leakage because of unequal expansion of boiler shell, tubes, and fire box. Portable fire-box boilers with return tubes are made in sizes as large as 150 horse power and for pressures as high as 150 pounds per square inch, but being more costly than some of the other types of boilers of equal capacity are used only where portability is an essential requirement.

62. Fitzgibbons Boiler. — Fig. 33 shows a section through a Fitzgibbons boiler and setting illustrating a combination of the vertical tubular and the locomotive fire-box type. This combination provides a large and efficient combustion chamber with economy of floor space. The horizontal tube sheets are completely submerged and the arrangement of the heating surface effects an exceedingly rapid water circulation. Fitzgibbons boilers are made in various sizes ranging from 10 to 350 horse power and are finding favor with engineers for small installations. They are much in evidence in public buildings and institutions.

63. Scotch-marine Boiler. — Where an internally fired boiler is desired for large powers the Scotch-marine type is finding much favor with engineers. A number of the tall office buildings in Chicago are equipped with boilers of this class which are giving good results. They require little overhead room, no brick setting, and are excellent steamers. The Continental boiler, Fig. 34, is one of the best known of this type. The boiler is self-contained and requires no brick setting, the only fire brick used being those that form the bridge wall, baffle ring, and the layer at the back of the combustion chamber. The furnace and tubes are entirely surrounded by water, so that all fire surfaces, excepting the

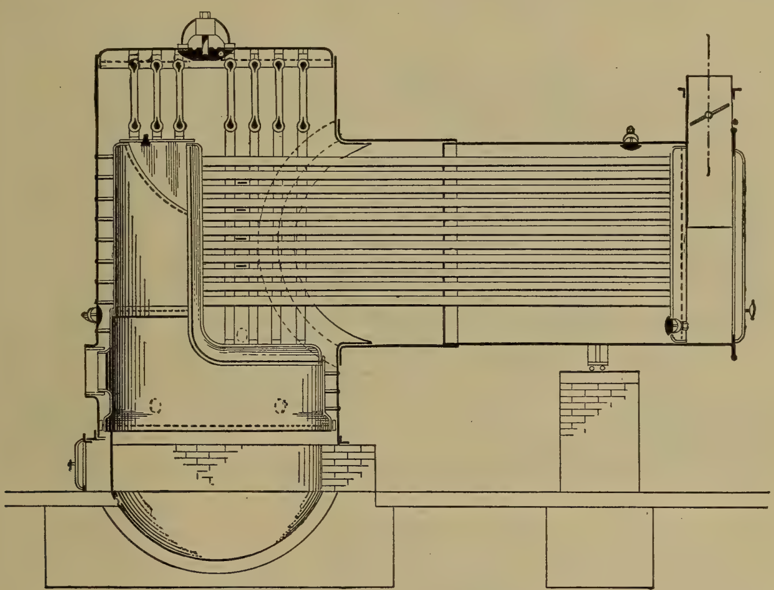


Fig. 33. 100-horse-power Fitzgibbons Boiler.

rear of the combustion chamber, are water cooled. The furnace is corrugated for its whole length. These corrugations, in addition to giving greater strength to the furnace, act as a series of expansion joints, taking up the strains due to unequal expansion of furnace and shell. Practically all types of mechanical stokers and grates are applicable to these boilers. The advantages of a Scotch boiler and of all internally fired boilers are: (1) minimum radiation losses; (2) requires no setting; (3) no leakage of cool air into the furnace as sometimes occurs through cracks or porous brickwork of other types; (4) large steaming capacity for the space occupied. The circulation, however, is not always positive and the water below the furnace may be considerably

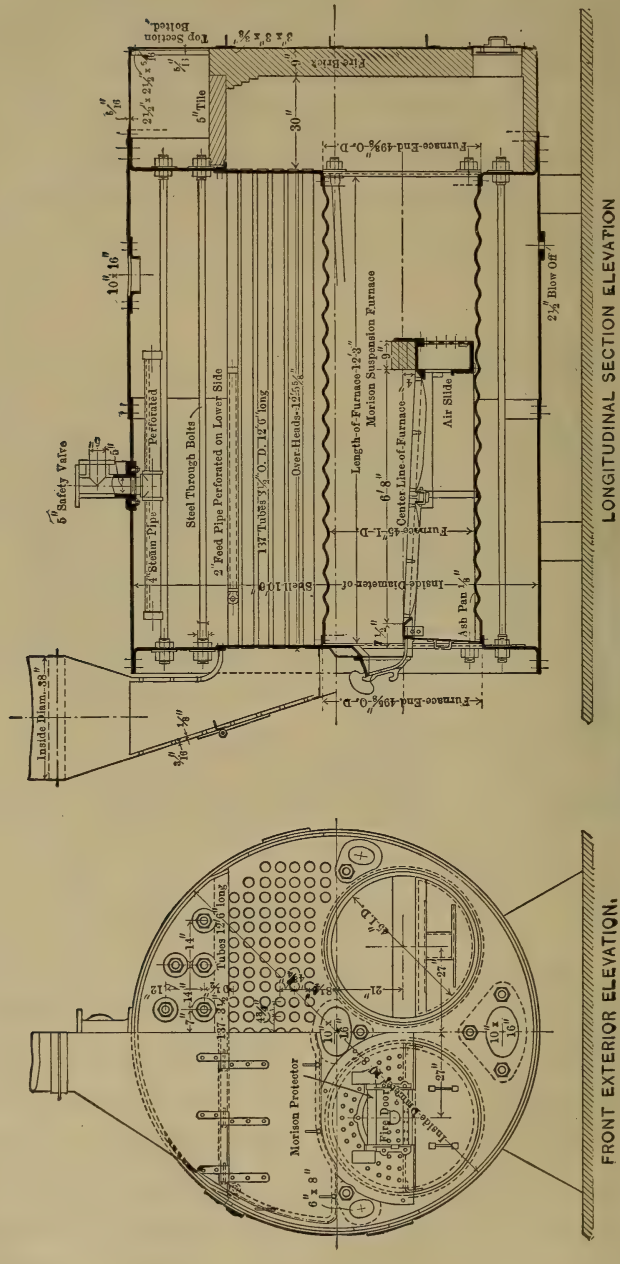


FIG. 34. Stationary Scotch-marine Boiler.

below the average or normal temperature, giving rise to unequal expansion and contraction which may cause leakage. The boiler proper is relatively costly, but this is offset to some extent by the absence of setting.

64. Robb-Mumford Boiler. — Fig. 35 shows a section through a Robb-Mumford boiler, which is a modification of the Scotch-marine and of the horizontal tubular type. It consists of two cylindrical shells, the lower one containing a round furnace and tubes and the upper one forming the steam drum, the two being connected by two necks. The lower shell has an incline of about one inch per foot from the horizontal, for the purpose of promoting circulation and draft,

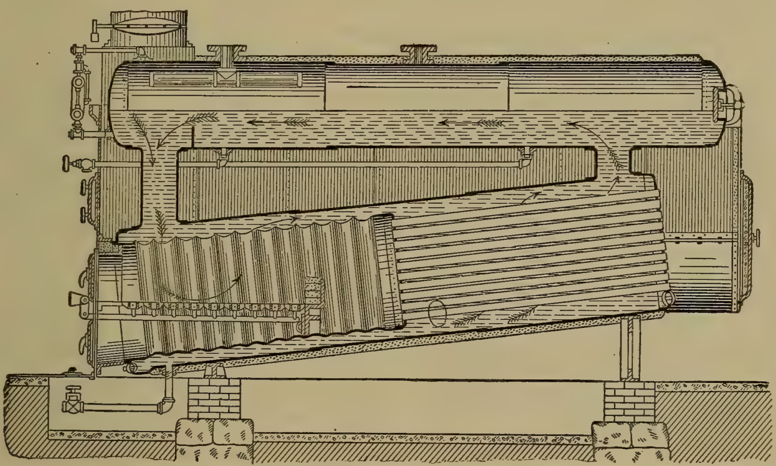


FIG. 35. Robb-Mumford Boiler.

and also for convenience in washing out the lower shell. Combustion takes place in the furnace, which is surrounded entirely by water, and the gases pass through the tubes and return between the lower and upper shells (this space being inclosed by a steel casing) to the outlet at the front of the boiler. Mingled water and steam circulate rapidly up the rear neck into the steam drum, where the steam is released, the water passing along the upper drum towards the front of the boiler and down the front neck, a semi-circular baffle plate around the furnace causing the down-flowing water to circulate to the lowest part of the lower shell under the furnace. The outer casing, which incloses the space between the lower and upper shells, including the rear smoke box and the smoke outlet, is constructed of steel plate, with angle-iron stiffeners, the various sections being bolted together for convenient removal. The inside of the steel case, including the rear smoke cham-

ber, is lined with asbestos air-cell blocks fitted in between the angle-iron stiffeners. The top of the upper drum and bottom of the lower shell are also covered with non-conducting material after the boiler is erected. Owing to the fact that steam and water spaces are divided between two cylindrical shells, the thickness of plates is not so great as in the Scotch-marine or horizontal return tubular types; and the rear chamber of the marine boiler is avoided.

The chief claim for this type of boiler is compactness. A battery of five 200-horse-power units occupies a floor space of but 33 feet in width by 20 feet in depth and 12.5 feet high. Each unit is entirely independent and may be isolated for cleaning, inspection, and repairs.

65. Horizontal Return Tubular Boilers. — These are the most common in use and are constructed in sizes up to 200 horse power. They are simple and inexpensive and, when properly operated, durable and economical. Figs. 36 to 39 show various forms of standard settings, and Figs. 85, 86, and 87 different "smokeless" settings. The grate is independent of the boiler, and the products of combustion pass beneath the shell to the back end, returning through the tubes to the front, and into the smoke connection.

The tubes are from 3 to 4 inches in diameter and from 14 to 18 feet long, and are expanded into the tube sheets. The portion of the tube sheets not supported by the tubes is secured against bulging by suitable stays. Access to the interior of the boiler is obtained through manholes. The most convenient arrangement for inspection and cleaning is to have one manhole located at the top of the shell and one at the bottom of the front tube sheet. Return tubular boilers are made either with an extended front (Fig. 36) or flush front (Fig. 37). The latter costs a little more for brick and setting, but it is more convenient to operate and the boiler is less expensive. The shell may be supported by lugs on the brickwork as in Fig. 36 or by steel beams and hangers as in Fig. 38. The latter construction permits the brickwork and shell to expand or contract independently, and settling of the brickwork does not affect the boiler alignment. With the side bracket support, the front lugs usually rest directly on iron or steel plates embedded in the brickwork, and the back lugs on rollers, to permit free expansion and contraction. The brackets are long enough to rest upon the outside wall, so that the inside brick lining can be renewed without disturbing the setting. The distance between the rear tube sheet and wall should be about 16 inches for boilers less than 60 inches in diameter and from 20 to 24 inches for larger ones. The distance between grate and boiler shell should not be less than 28 inches for anthracite coal and 36 inches

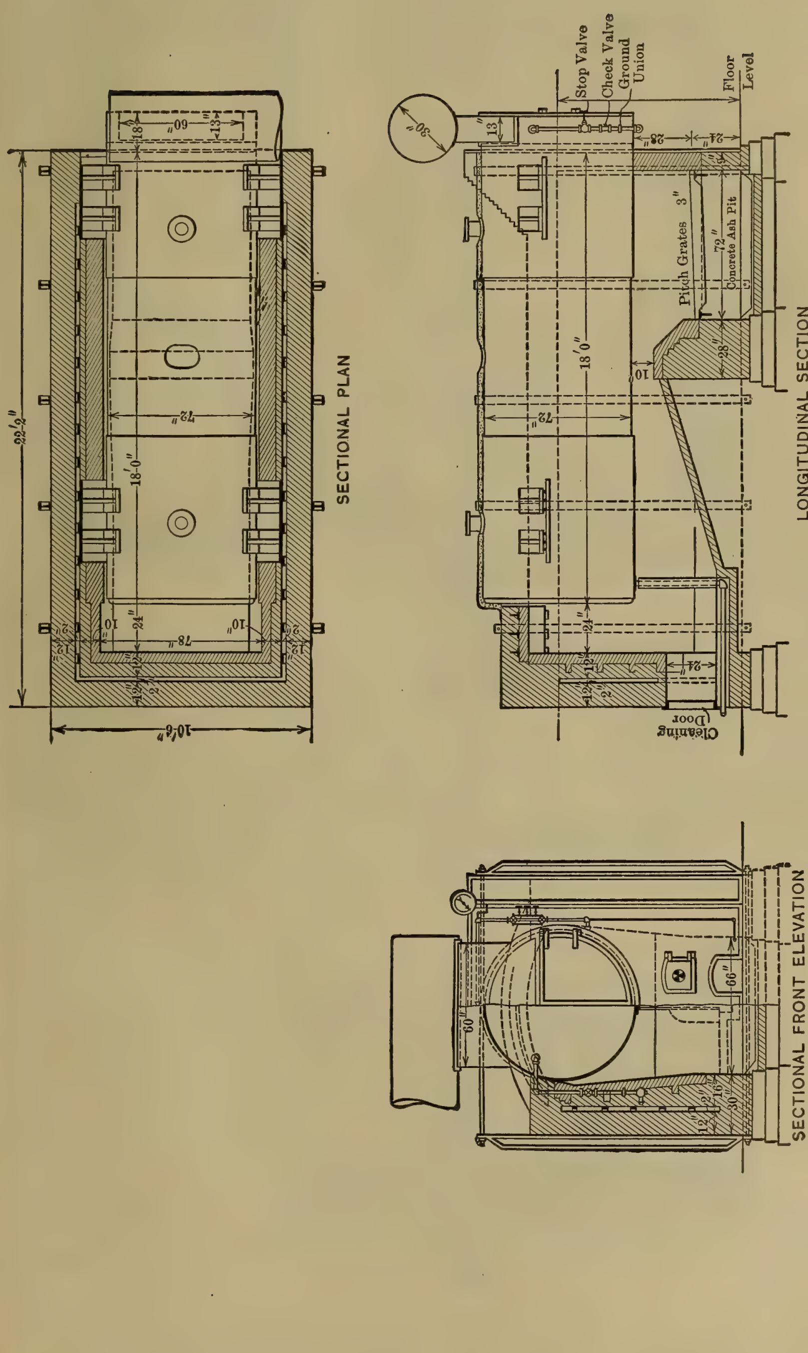


FIG. 36. Return Tubular Boiler Setting — Extended Front.

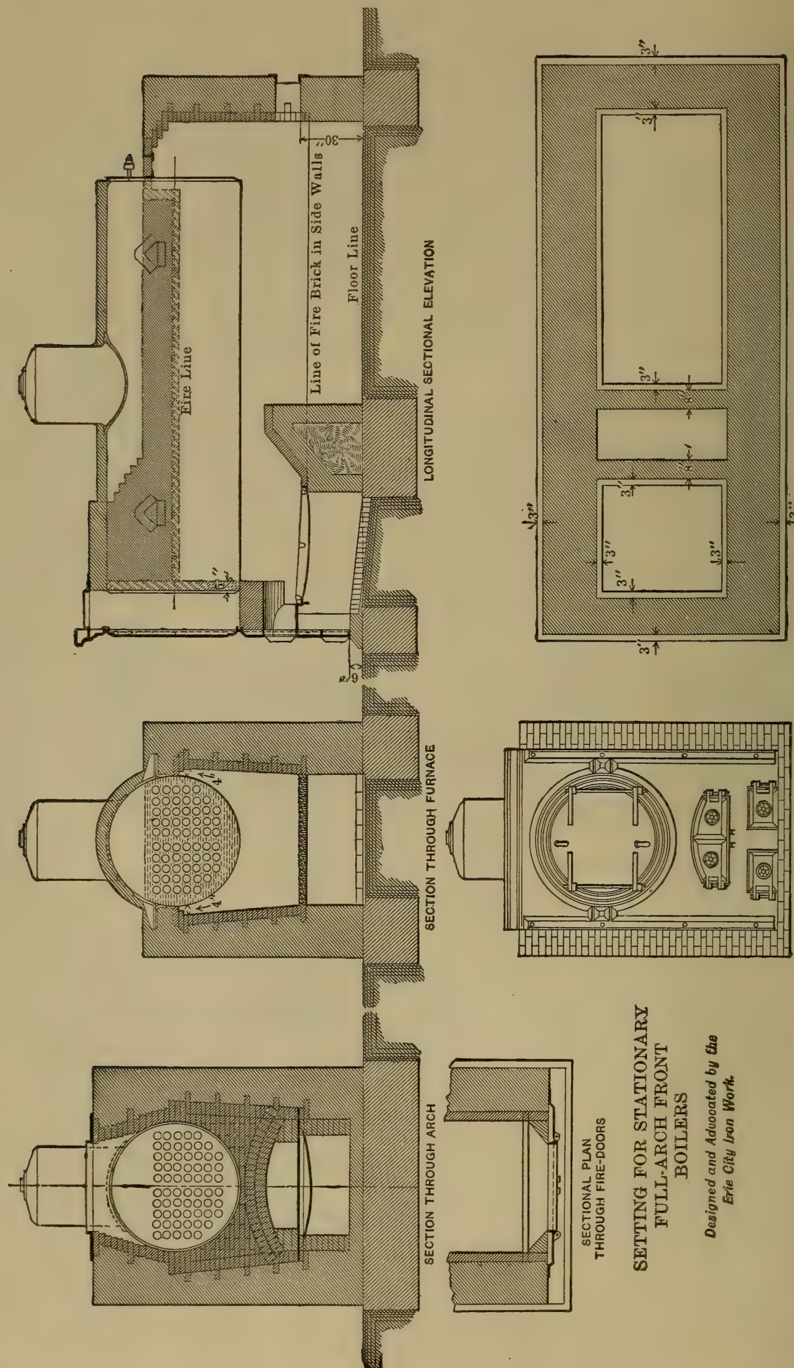


FIG. 37. Return Tubular Boiler Setting. — Flush Front.

for bituminous coal.* The greater this distance the more complete the combustion, since the gases will have a better opportunity for combining with the air before coming into contact with the comparatively cool surfaces of the shell. The shell should be slightly inclined toward the blow-off end so as to drain freely.

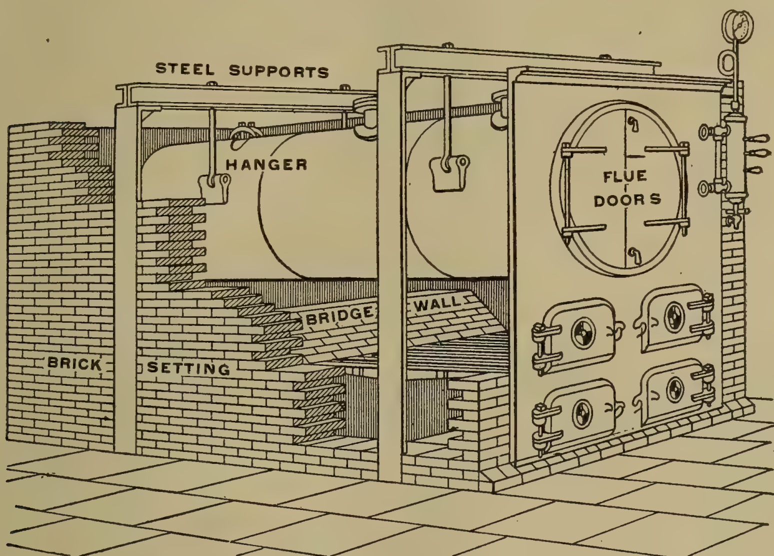


Fig. 38. Return Tubular Boiler Setting — Steel Beam Suspension.

The vertical distance between the bridge wall and shell is usually between 10 and 12 inches. The lower part of the combustion chamber behind the bridge wall may be filled with earth and paved with common brick as in Fig. 39 or left empty as in Fig. 37. The shape of the bridge walls whether curved to conform to the shell or flat appears to have little influence on the economy.

The side and end walls are ordinarily constructed of common brick with an inner lining of fire brick, and may be solid as in Fig. 37 or double with air spaces as in Fig. 36.† The latter construction is preferable and permits the inner and outer walls to expand independently without cracking and settling. The side walls are braced by five pairs of buckstaves, with through rods under the paving and over the tops of the boilers.

* For smokeless combustion the setting must be modified. See furnaces illustrated and described in paragraph 97.

† See "The Flow of Heat through Furnace Walls," Bulletin No. 8, U. S. Bureau of Mines, 1911.

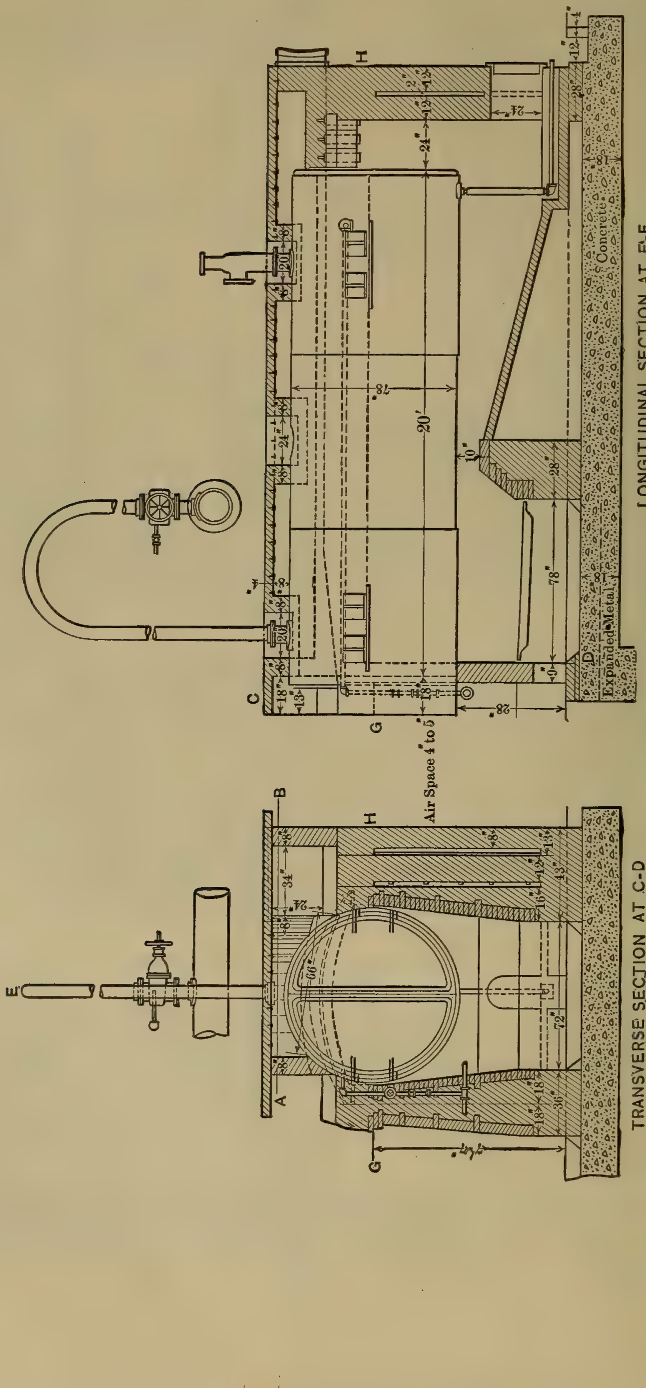


Fig. 39. Boiler Setting; "Wood" Mill of the American Woolen Company, Lawrence, Mass.

The connection between the rear wall and the shell is a source of more or less trouble on account of the expansion and contraction of the boiler. Cast-iron supports of T section supporting a fire-brick arch are usually employed as illustrated in Fig. 40, the clearance between the arch and the shell being sufficient to allow the necessary expansion.

Fig. 41 shows the common method of resting one end of the arch supports on the rear wall and the other end on an angle iron riveted to the boiler.

The products of combustion are sometimes carried over the top of the boiler as shown in Fig. 39. This tends to superheat the steam, but the advantage gained is probably offset considerably by the extra cost of the setting and the accumulation of soot on the top of the shell. The arrangement is not common.

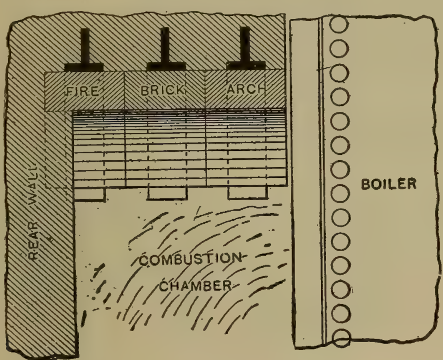


Fig. 40. Furnace Arch Bars.

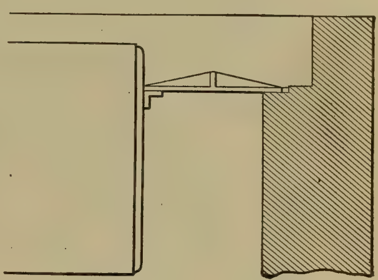


Fig. 41. Back Connection made with Cast-iron Plate.

The steam connection is naturally made to the highest point in the boiler shell. Frequently a steam dome, to which the steam nozzle is connected, is provided as in Fig. 37. The function of the steam dome is to increase the steam space so as to permit the collection of dry steam at a point high above the water level. If a boiler is too small for its work and is forced far above its rating a steam dome is probably an advantage, though its use is less common now than formerly, since a properly designed boiler insures ample steam space without one. A dry pipe inside the boiler above the water line as in Fig. 34 or 35 is commonly used to guard against priming where the nozzle is connected to the shell.

For low pressures and small powers the return tubular boiler has the advantage of affording a large heating surface in a small space and large overload capacity. It requires little overhead room and its first cost is low. On the other hand the interior is difficult of access for purposes

of cleaning and inspection. Boilers of this type are constructed in various sizes ranging from a 36″ × 8′, rated at 15 horse power, to a 96″ × 21′, rated at 400 horse power, though sizes above 200 horse power are exceptional. The working pressure seldom exceeds 150 pounds per square inch.

66. Lyons Boiler. — The standard externally fired return tubular boiler is limited in size since the danger from overheating the shell directly over the fire bed increases rapidly with the increase in thickness of the plate. The Lyons boiler, a section through which is shown

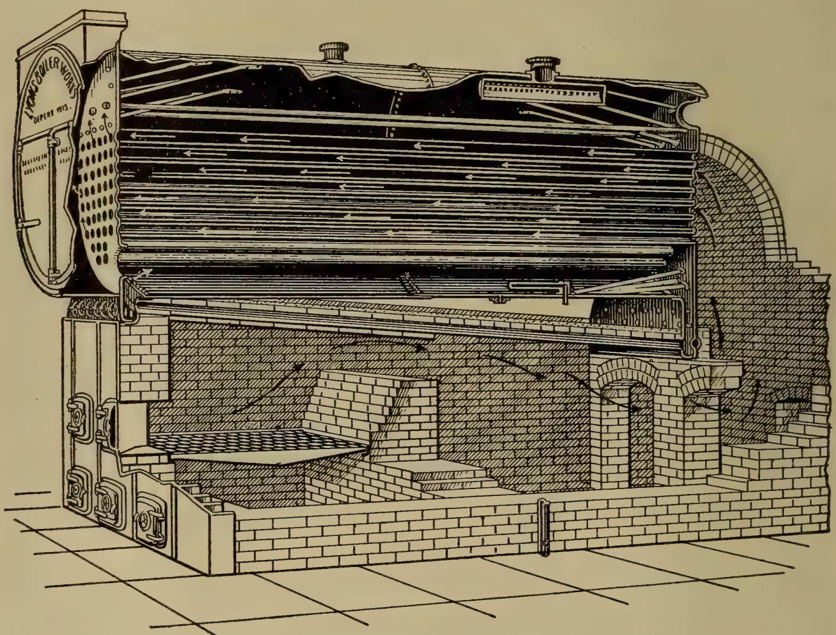


FIG. 42. Lyons Boiler and Setting.

in Fig. 42, overcomes this restriction through the addition of a bank of water tubes which form a roof to the furnace. These tubes protect the shell from the direct action of the gases and insure a positive and rapid circulation. They are covered with tile or split brick and form the equivalent of a "Dutch oven." Lyons boilers are made in various sizes up to 450 horse power.

67. Sederholm Boiler. — Fig. 43 shows a longitudinal section and a vertical sectional elevation of a Sederholm boiler with stationary grate and setting. This boiler is a modification of the old "elephant" type so much in evidence in France. As will be seen from the illustration it consists essentially of a return tubular boiler fitted with four hori-

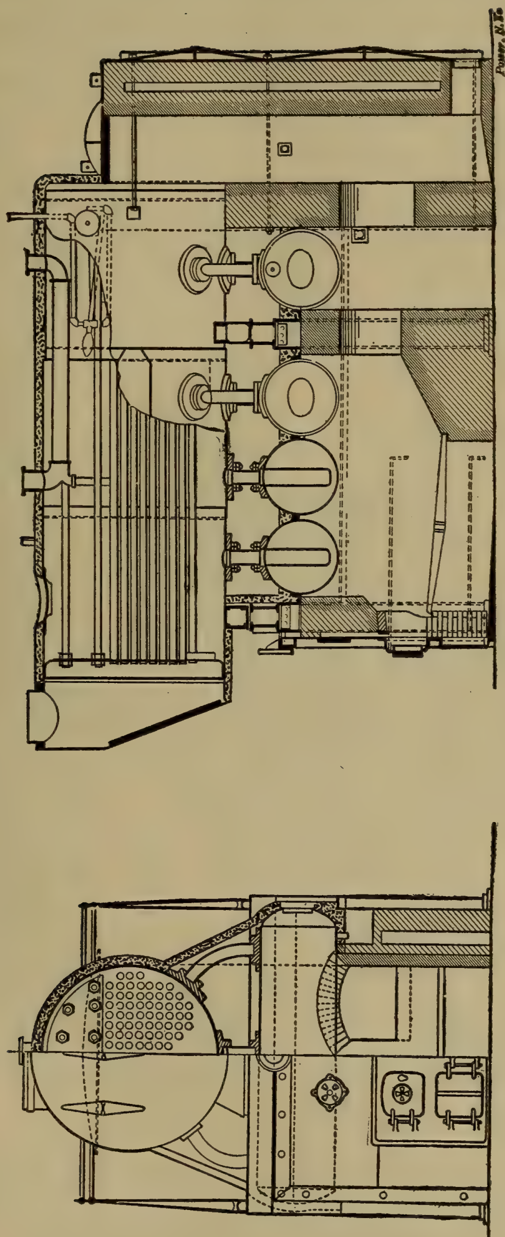


FIG. 43. Sederholm Boiler and Setting.

zontal water drums extending the full width of the setting. The drums form a roof for the furnace and protect the bottom of the shell from direct flame impingement. A large portion of the impurities in the feed water is precipitated in the drums from which it is readily

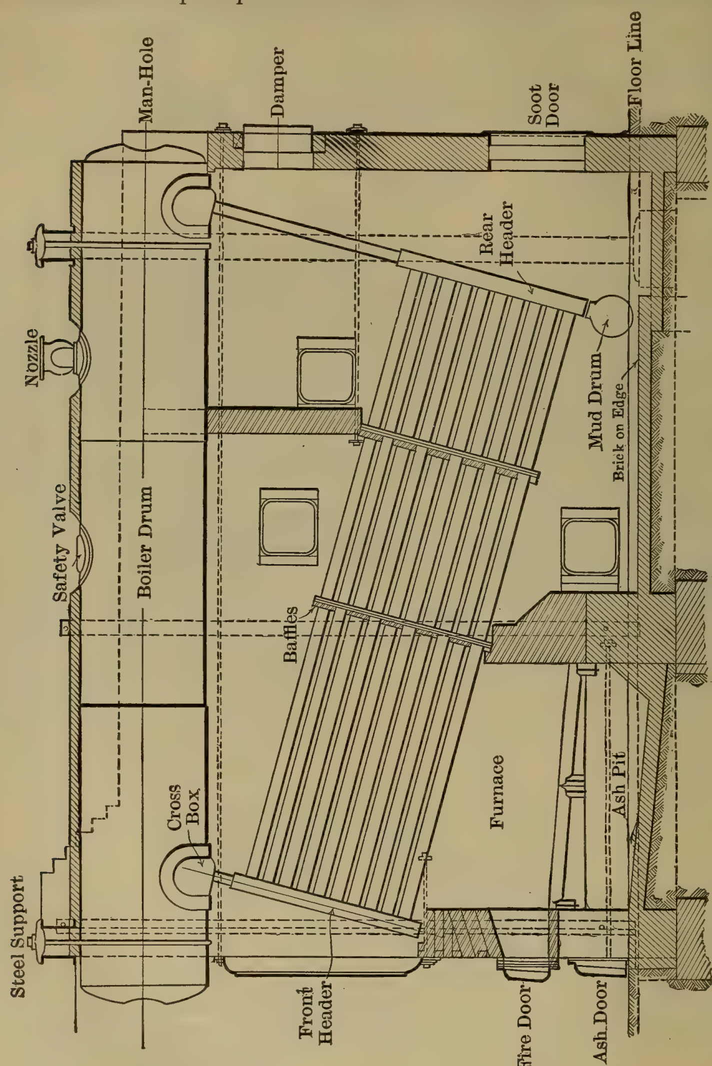


FIG. 44. Babcock and Wilcox Boiler and Setting.

discharged. A number of Sederholm boilers equipped with chain-grate stokers are installed in the power plant of the Commercial National Bank Building, Chicago, and are giving excellent results.

68. Babcock and Wilcox Boiler. — Fig. 44 shows a longitudinal section through a Babcock and Wilcox boiler, illustrating a typical horizontal

water-tube type. The tubes, usually 4 inches in diameter and 18 feet in length, are arranged in vertical and horizontal rows and are expanded into pressed-steel headers. Two vertical rows are fitted to each header and are "staggered" as shown in Fig. 45. The headers are connected with the steam drum by short tubes expanded into bored holes. Each tube is accessible for cleaning through openings closed by covers with ground joints held in place by wrought-iron clamps and bolts. The tubes are inclined at an angle of about 22 degrees with the horizontal.

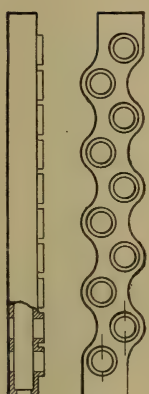


Fig. 45. — Details of Header — Babcock and Wilcox Boiler.

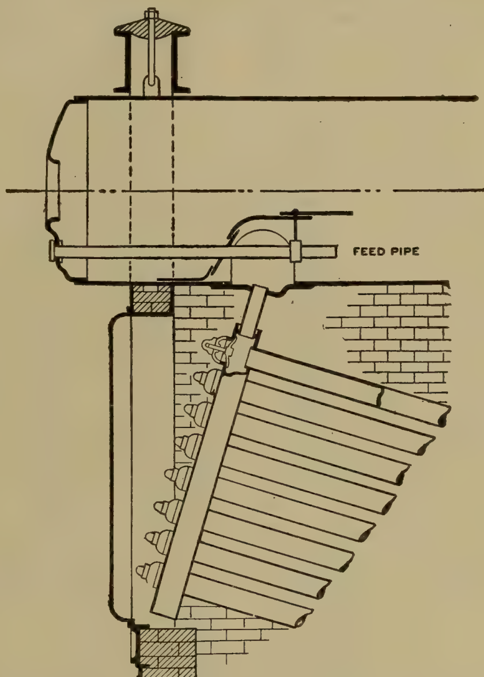


Fig. 46. Front Section — Babcock and Wilcox Boiler.

The rear headers are connected at the bottom to a cast-iron mud drum. The steam drum is horizontal and the headers are arranged either vertically or at right angles to the tubes. The boiler is supported by steel girders resting on suitable columns independent of the brick setting. The grate is placed under the higher ends of the tubes, the products of combustion passing at right angles to the tubes and being deflected back and forth by fire-tile baffles. The feed water enters the front of the steam drum as shown in Fig. 46. A rapid circulation is effected by the difference in density between the solid column of water in the rear header and the mixed steam and water in the front one. B. & W.

boilers under 150 horse power have but one steam drum, and the larger sizes have two. The number of tubes varies with the size of boiler,

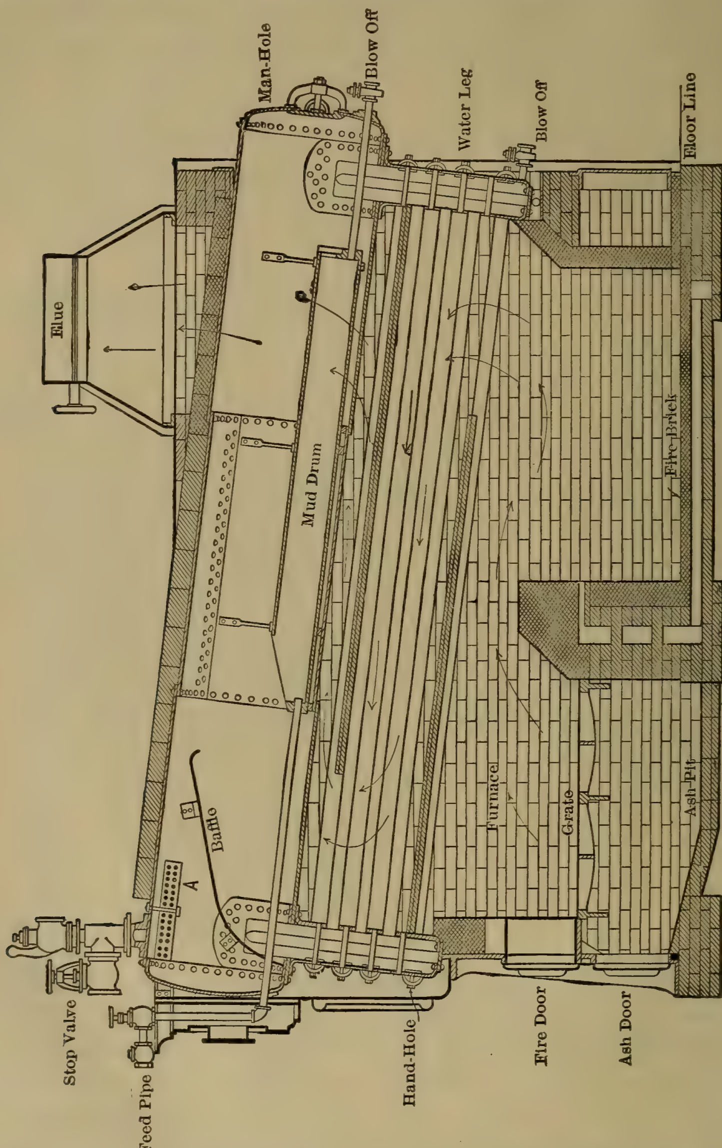


Fig. 47. Heine Boiler and Setting.

ranging from 6 wide and 9 high in the 100-horse-power boiler to 14 high and 18 wide in the 500-horse-power boilers.

69. Heine Boiler. — Fig. 47 shows a longitudinal section through a Heine horizontal water-tube boiler. This boiler differs from the B. & W.

boiler in that the tubes are expanded into a single large header constructed of boiler steel. The drum and tubes are parallel with each other and inclined about 22 degrees with the horizontal. The feed water enters at the front of the steam drum and flows into the mud drum, from which it passes to the rear header. Steam is taken from the front of the steam drum and is partially freed from moisture by the dry pipe *A*. A baffle over the front header prevents an excess of water from being carried into the dry pipe. As the rear header forms one large chamber, no additional mud drum is necessary and the sediment is blown off from the bottom by the blow-off cock. The circulation is somewhat freer than in the B. & W. boiler on account of the large sectional area through the headers.

70. Wickes Boiler. — Fig. 48 shows a section through a Wickes vertical boiler, illustrating the vertical water-tube type. The steam drum and water drum are arranged one directly above the other. The tubes are expanded and rolled into both tube sheets and are divided into two sections by fire-brick tile. The water line in the steam drum is carried about two feet above the tube sheet, leaving a space of five feet between water line and top of the drum. This affords a large steam space and disengagement surface. Feed water is introduced into the steam drum below the water line and flows downward through the tubes of the second compartment. The boiler is supported by four brackets riveted to the shell of the bottom drum and is independent of the setting. The entire boiler is inclosed in brickwork and is completely surrounded by the products of combustion. The upper part of the steam drum acts as a superheating surface and tends to dry the steam. Wickes boilers are simple in design, easy to inspect and clean, low in first cost, and comparable in efficiency with any water-tube type of boiler.

71. Parker Boiler. — Fig. 49 shows a longitudinal sectional elevation and an end sectional elevation of a 1200-horse-power Parker down-flow boiler with double-ended setting. This type of boiler is finding much favor with engineers for central stations where large units are desired. The Parker boiler differs from the conventional horizontal water-tube boiler principally in circulation and flexibility.

Feed water is pumped into the economizer or feed element (1), Fig. 49, at *O*, *O*, and flows downward through a series of tubes, discharging finally into the drum through an upcast *H*. In a large unit, as illustrated here, there are two feed elements and two drums. The circulation in the feed element is indicated by solid lines and arrow points at the left of the end sectional elevation, the tubes having been omitted from the drawing for the sake of clearness.

The intermediate elements (2) take their water supply from the bottom of the drum through a cross-box V, the circulation being downward, as indicated by arrow points, through four tube wide elements, and finally discharge it through an upcast X into the steam space of the drum.

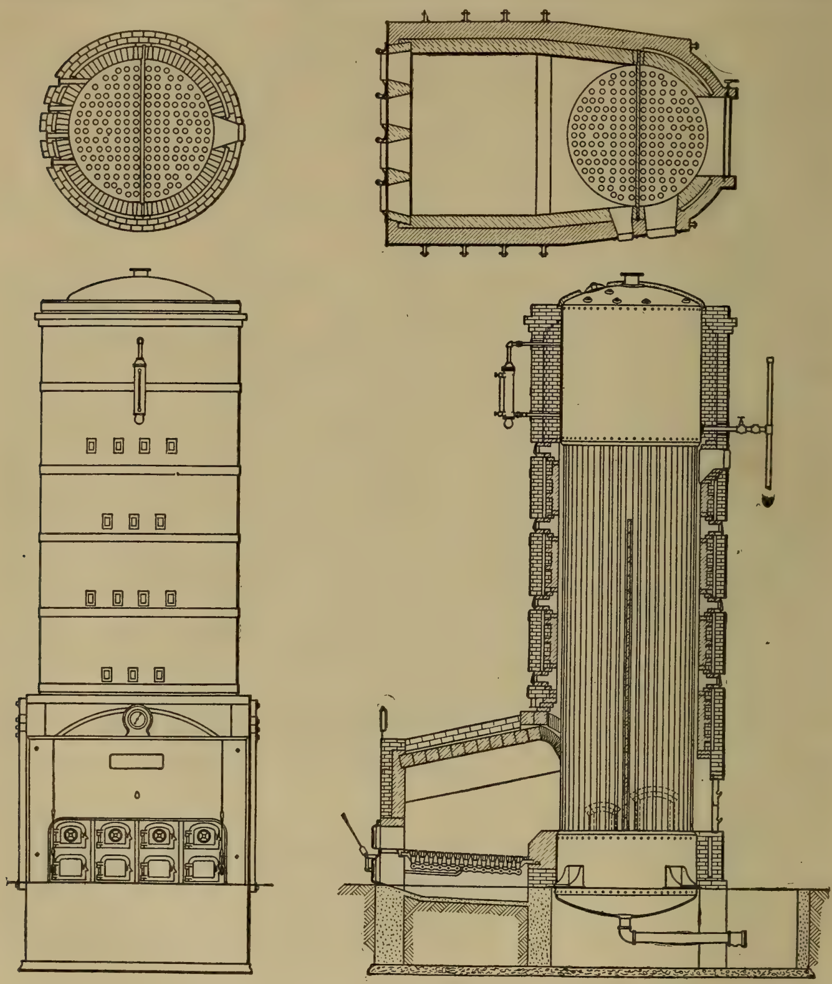


Fig. 48. Wickes Vertical Water-tube Boiler.

Each element has a "down-comer" and an upcast. In the smaller-sized boilers the intermediate elements are omitted.

The evaporator elements (3) take their water supply from the bottom of the drum at V, the circulation being downwards through two tube wide elements, and finally discharge it into the drum at U. The last two passes of the water are through the two bottom tubes of each

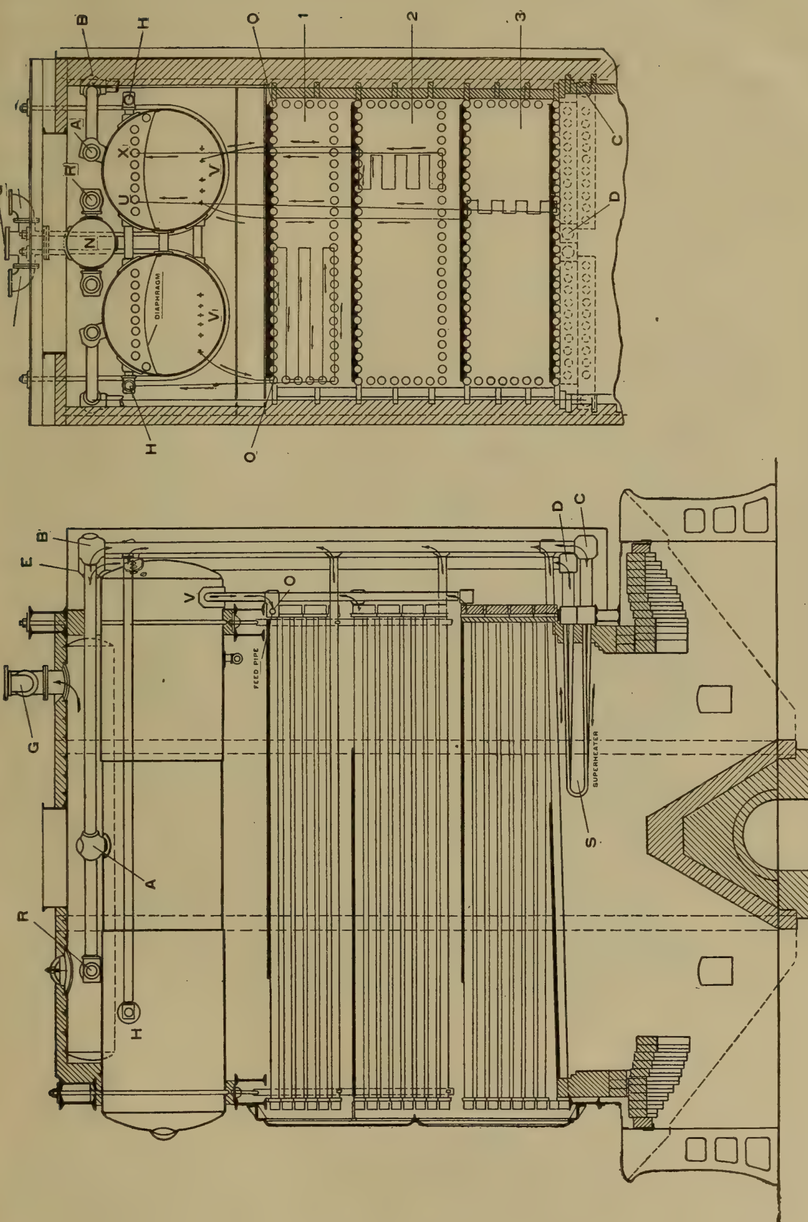


FIG. 49. 1200-H.P. Parker Down-Flow Boiler with Double-Ended Setting.

element, thus assuring dry steam without the use of dry pipes. To prevent reversal of flow each element is fitted with a check valve at the admission end. Each drum is equipped with a diaphragm, as indicated, separating the steam and water spaces, thus insuring against foaming and priming.

Saturated steam is taken from the drum at *A* and passes by way of *B* to *C*, where it enters the superheater *S*. The superheated steam leaves the superheater at *D* and passes by way of *E* and *R* to the storage drum *N*, finally leaving the boiler at *G*. The superheater is designed to maintain an approximately constant degree of superheat for all variations in load.

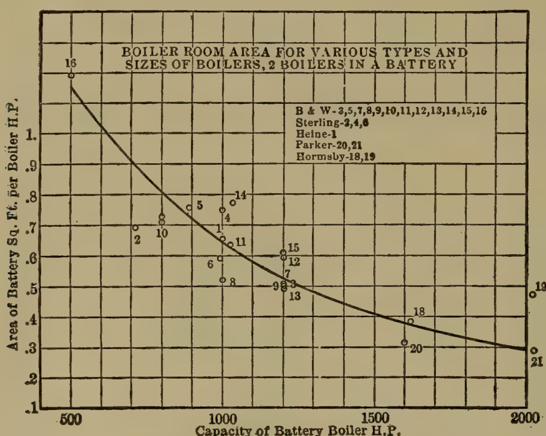


FIG. 50.

All tubes are connected by malleable-iron junction boxes the interior of each tube being accessible through hand holes placed opposite the end of each tube. The hand-hole cover plates are on the inside of the box and have conical ground joints, thus dispensing with gaskets.

The Parker boiler is built single or double ended, with or without superheater, and in sizes ranging from 50-horse-power to 2500-horse-power standard rating.

72. Stirling Boiler. — Fig. 51 shows a longitudinal section through a Stirling water-tube boiler, which differs considerably from the types just described. Three horizontal steam drums and one horizontal mud drum are connected by a series of inclined tubes. The tubes are bent at the ends to permit them to enter the drums radially. Short tubes connect the steam spaces of all the upper drums and also the water spaces of the front and middle drums. Suitably disposed fire-tile baffles between the banks of tubes direct the gases in their proper course. The boiler is supported on a structural steel framework independent of the setting. The feed water enters the rear upper drum, which is the cooler part of the boiler, and flows to the bottom or mud drum, where it is heated to such an extent that many of the impurities are precipitated. There is a rapid circulation up the front bank of tubes to the front drum, across to the middle drum, and thence down

the middle bank of tubes to the mud drum. The interior of the drums is accessible for cleaning by manholes located in the ends. The Stirling furnace is distinctive in design. A fire-brick arch is sprung over the grates immediately in front of the first bank of tubes. The large triangular space between boiler front, tubes, and mud drum forms the combustion chamber. Stirling boilers are somewhat lower in first cost than other types of water-tube boilers on account of the absence

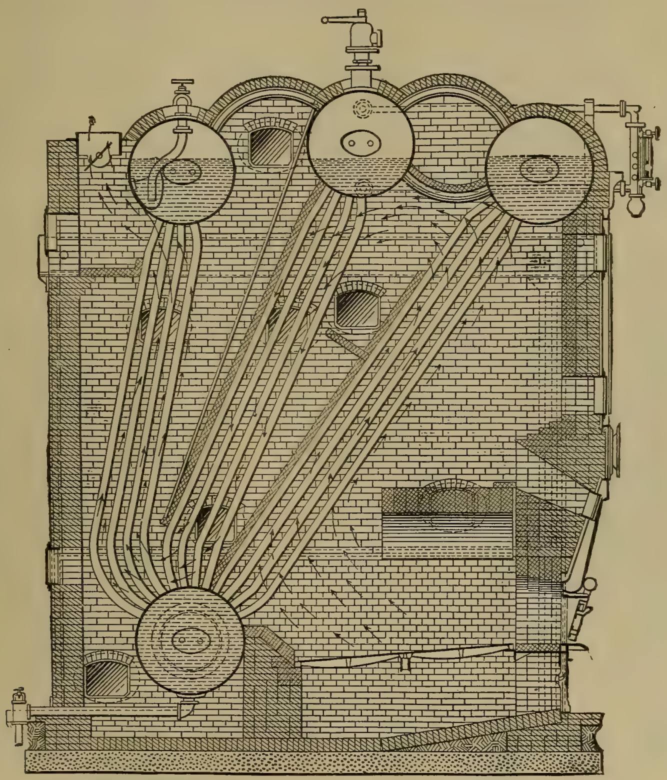


FIG. 51. Stirling Boiler and Setting.

of numerous hand holes and the like which are necessary in the horizontal type.

Fig. 52 gives a sectional view through the boiler and setting of a 2365-horse-power Stirling boiler equipped with Taylor stokers as installed at the Delray station of the Detroit Edison Company. Five boilers are now in operation and it is planned to eventually install ten. Though rated at 2365 boiler horse power they are capable of carrying continuously a load equivalent to 6000 kilowatts with a maximum of 8000 kilowatts. The overall dimensions of the boiler and

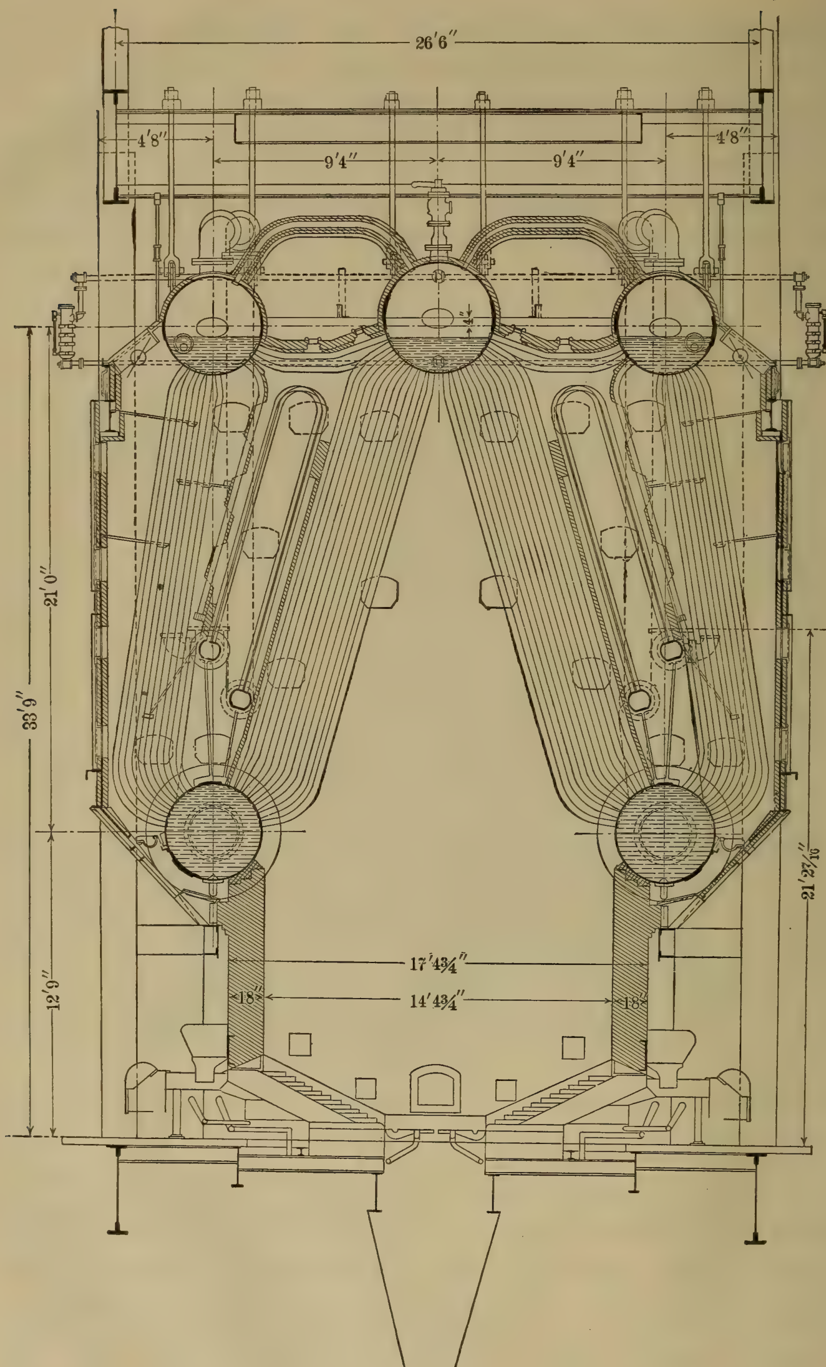


Fig. 52. 2365-horse-power Stirling Boiler — Delray Station, Detroit Edison Company.

setting are shown in the illustration. Each unit contains 23,654 square feet of effective heating surface and is provided with superheaters for supplying steam at 150 degrees superheat. Table 31 gives a résumé of the principal results obtained from tests of these units with Roney and Taylor stokers. The grate surface per boiler for the Roney stoker

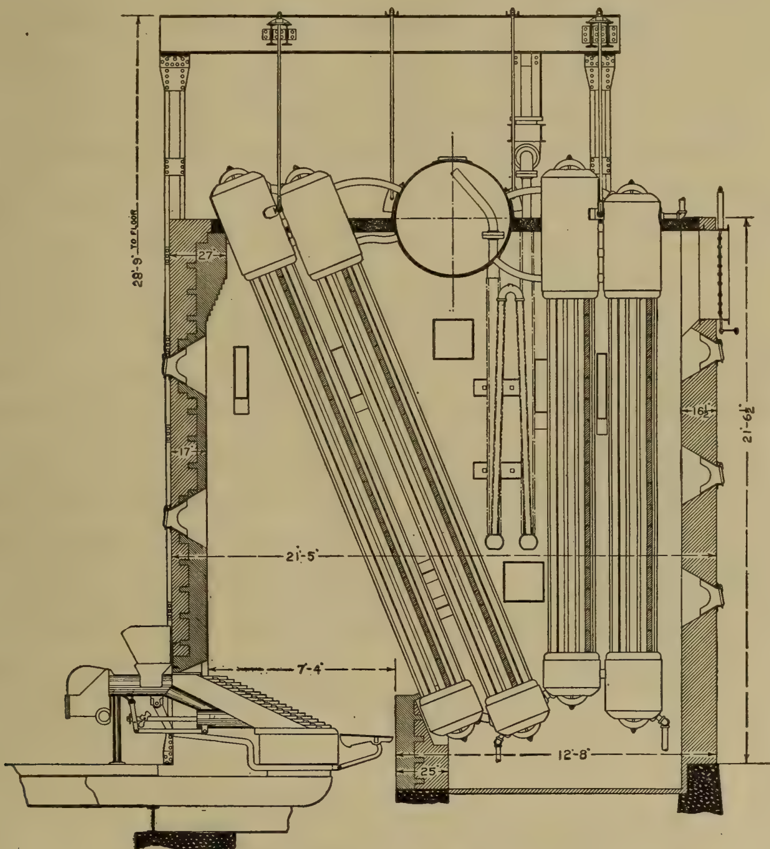


FIG. 53. Bigelow-Hornsby Boiler and Setting.

is 446 square feet and for the Taylor stoker 405 square feet, thus giving as ratios of grate surface to heating surface 1 : 53 and 1: 58.5 respectively. For a complete description of these tests see Jour. A.S.M.E., Nov., 1911, p. 1439.

73. The Bigelow-Hornsby Boiler. — Fig. 53 shows a vertical section through a Bigelow-Hornsby boiler equipped with Foster superheater and Taylor stoker. This boiler is of the vertical water-tube type and is made up of a number of cylindrical elements, each element comprising

an upper and lower drum connected by straight tubes. The two front elements are inclined over the furnace at an angle of about 68 degrees, and the two rear elements are vertical. The upper drums of the elements are connected to a horizontal main steam drum by flexible tubing as indicated. Four elements constitute a section with an effective heating surface of 1250 square feet. Any number of sections may be connected together forming units of from 250 to 2500 boiler horse power or more. All parts, both external and internal, are readily accessible. Feed water enters the top drum of the rear elements and passes twice the length of the tubes before entering into the general circulation. This arrangement permits a considerable portion of the impurities in the water to be precipitated in the rear drum from which they are readily discharged. By the time the water reaches the front of the boiler directly over the furnace, where the heat transmission is the most intense, the scale-forming elements have been practically eliminated. The particular features of this boiler lie in the great extent of heating surface exposed to radiant heat and the height and volume of the combustion chamber. Bigelow boilers are productive of high economy and are readily forced to twice their rated capacity with little decrease in over-all efficiency. The most notable installation of Bigelow boilers in this country is at the power plant of the Hartford Electric Light & Power Company, Hartford, Conn., where two 1250- and one 2500-boiler-horse-power units are installed. The latter is the largest single boiler and setting in the world at this writing (Feb., 1912).

74. Unit of Evaporation. — The performance of a boiler and furnace may be expressed in terms of the weight of water evaporated per hour per square foot of heating surface or of the weight evaporated per pound of fuel. To reduce all performances to an equal basis so as to facilitate comparison the evaporation under actual conditions is conveniently referred to the equivalent evaporation from a feed-water temperature of 212 degrees F. to steam at atmospheric pressure. The heat required to evaporate one pound of feed water at a temperature of 212 degrees F. into steam of the same temperature, or "from and at 212 degrees" as it is commonly called, is 970.4 B.t.u. The ratio of the heat necessary to evaporate one pound of water under actual conditions of feed temperature and steam pressure to the heat required to evaporate one pound from and at 212 degrees is called the *factor of evaporation*. Thus, for dry steam,

$$F = \frac{\lambda - q_2^*}{970.4}, \tag{27}$$

* For most practical purposes q_2 may be taken as $t - 32$, in which t = temperature of the feed water, degrees F.

in which

F = factor of evaporation,

λ = total heat of one pound of steam at observed pressure above 32 degrees F.,

q_2 = total heat of one pound of feed water above 32 degrees F.

If the steam is wet,

$$\lambda = xr + q, \tag{28}$$

in which

x = the quality of the steam,

r = latent heat of evaporation at observed pressure,

q = heat in liquid at observed pressure.

If the steam is superheated,

$$\lambda = r + q + Ct_s, \tag{29}$$

in which

C = the mean specific heat of the superheated steam,

t_s = the degree of superheat, degrees F.

75. Heat Transmission. — Fig. 54 shows a section through a boiler-heating plate and serves to illustrate the accepted theory of heat transmission. The outer surface of the plate is covered with a thin layer of soot and a film of gas, and the inner surface is similarly protected by a layer of scale and a film of steam and water. It is, therefore, reasonable to assume that the *dry* surface of the plate is located somewhere within the film of gas, and the *wet* surface within the film of water and steam.

The heat is imparted to the *dry* surface by: (1) *radiation* from the hot fuel bed and furnace walls, and by (2) *convection* from the moving furnace gases. The heat is transferred through the boiler plate and its coatings purely by *conduction*. The final transfer from the *wet* surface to the boiler is mainly by *convection*.

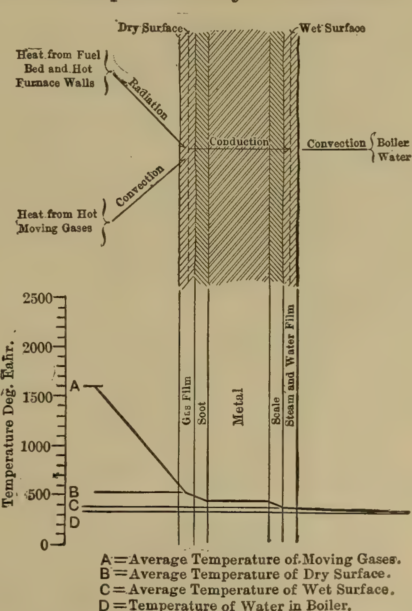


Fig. 54. — Heat Transmission through Boiler Plate.

Radiation depends on the temperature, and according to the law of Stefan and Boltzmann is approximately proportional to the difference

between the fourth power of the absolute temperature of the fuel bed and furnace walls and the temperature of the dry surface of the heating plate. According to this law the heat transmitted by radiation increases rapidly with the increase in furnace temperature. In the ordinary boiler and setting the surface exposed to radiation is only a small portion of the total heating surface, and, since in well-operated furnaces the temperature of the furnace cannot be increased materially on account of practical considerations, there is little hope of increasing the capacity of a boiler by increasing the furnace temperature. The *extent* of heating surface exposed to radiation, however, may be greatly increased. Many authorities are of the opinion that the boiler of the future will depend largely upon radiation.

The heat imparted to a boiler plate by *convection* may be determined by the following equation (Prof. Perry, "The Steam Engine," 1906 Ed., p. 588):

$$H = C\,(t_1 - t_2)\,vd, \tag{30}$$

in which

H = B.t.u. transferred per hour per sq. ft. of heating surface,
C = a coefficient determined by experiment,
t_1 = temperature of the moving gases, degrees F.,
t_2 = temperature of the dry plate surface, degrees F.,
v = velocity of the gases, feet per second,
d = density of the gases, pounds per cubic foot.

Professor Nicholson gives the following modifications of formula (30) as applied to boiler tubes or flues (Engr. Lond., Feb. 19, 1908):

$$H = \left\{\frac{t}{200} + \frac{1}{40\,t}\left(1 + \frac{1}{m}\right)\right\}(t_1 - t_2)\,vd, \tag{31}$$

in which

t = mean film temperature,
m = hydraulic radius = area of tube in square inches ÷ perimeter of the tube in inches; other notations as in (30).

Both equations are based upon the same general law except that the latter gives a means of determining coefficient C in terms of the mean film temperature and the dimensions of the flues or tubes.

An examination of equation (30) shows that for a given set of conditions the heat imparted by convection to a unit of dry surface of heating plate varies directly as the difference between the temperature of the hot gases and that of the dry surface and directly as the velocity and density of the gases. However, the density of the gases drops with the rise of temperature, and increase in furnace temperature does not necessarily imply increase in heat impartation. It is the utiliza-

tion of the *velocity* factor, then, which offers a possibility of increasing boiler capacity.

Experiments by Professor Nicholson and the U. S. Geological Survey show that by establishing a powerful scrubbing action between the gases and the boiler plate the protecting film of gas is torn off as rapidly as it is formed and new portions of the hot gases are brought into contact with the plate, thereby greatly increasing the rate of heat transmission. Similarly, the faster the circulation of the water the greater will be the scrubbing action tending to remove the bubbles of steam from the wet surface and the more rapid will be the transfer from the plate to the boiler water.

The resistance of the metal itself is so small that it may be neglected in calculating the heat transmission, and it may be logically assumed that the plate will take care of all the heat that reaches its dry surface.

Professor Nicholson found that by filling up the flue of a Cornish boiler with an internal water vessel, leaving an annular space of only 1 inch around the latter, an evaporation eight times the ordinary rate was effected at a flow of gases 330 feet per second (8 to 10 times the average flow). The fan for creating the draft consumed about $4\frac{1}{2}$ per cent of the total power.

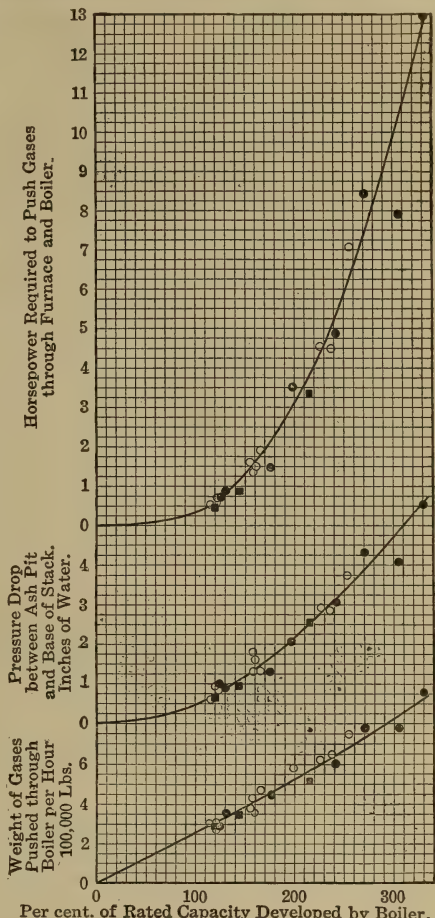


FIG. 55. Influence of Draft on the Capacity of a Normand Water-tube Boiler on the U. S. Torpedo Boat "Biddle."

The conclusion is that the heating surface for a given evaporation at the present rating may be reduced as much as 90 per cent for the same output, with a corresponding reduction in the size, cost, and space requirements, or with a given heating surface of standard rating the output may be enormously increased; also the increase in power necessary

to create the draft is by no means comparable with the advantages gained.

The modern locomotive boiler is the nearest approach to these conditions in practice. Here a powerful draft forces the heated gases through small tubes at a very high velocity and an enormous evaporation is effected with a comparatively small heating surface. See Fig. 55 for influence of draft on the capacity of a torpedo-boat boiler (Power and Engr., May 24, 1910).

These principles have been applied to a limited extent to stationary boilers already installed by making the gas passages smaller as compared to the length by means of suitable baffles (Fig. 49) and by forcing larger weights of gas through the boiler, either by forced draft or by increasing the grate area (Fig. 52).

In a general sense when the capacity of a boiler is doubled or tripled the over-all efficiency of the whole steam-generating apparatus drops, but the advantage gained usually offsets the loss in fuel economy. A close examination of the results, however, will show that the loss in efficiency is due more to low furnace efficiency than to inability of the boiler to absorb the heat generated.

In view of recent experiments it is not unlikely that within the next ten years boilers will be constructed capable of developing a boiler horse power with two or three square feet of heating surface instead of ten square feet, as at present, and with high over-all efficiency. (See Figs. 57 and 58.)

Heat Transmission in Boilers, Kreisinger and Ray: Power and Engr., June 29, 1909, p. 1144; Bulletin No. 18 U. S. Bureau of Mines, 1912; Journ. West Soc. Engrs., Sept. 18, 1907; Am. Inst. Elect. Engrs., Dec. 13, 1907.

Heat Transfer and Future Boiler Practice: A. H. Allen, Power and Engr., Sept. 21, 1909, p. 482; Engng., Lond., Feb. 19, 1908.

The Heat of Fuels and Furnace Efficiency: W. D. Ennis, Power and Engr., July 14, 1908, p. 50.

A Study in Heat Transmission (The Transmission of Heat to Water in Tubes as Affected by the Velocity of the Water), J. K. Clement and C. M. Garland, Univ. of Ill. Bulletin No. 40, Sept. 27, 1909; Power & Engr., Feb. 7, 1911, p. 222.

76. Heating Surface. — All parts of the boiler shell, flues, or tubes which are covered by water and exposed to hot gases constitute the heating surface. Any surface having steam on one side and exposed to hot gases on the other is superheating surface. According to the recommendations of the American Society of Mechanical Engineers, the side next to the gases is to be used in measuring the extent of the heating surface. Thus measurements are made of the inside area of fire tubes and the outside area of water tubes. The heating surface in

a boiler under average conditions of good practice is most efficient when the heated gases leave the uptake at a temperature of 100 to 200 degrees F. above that of the steam. Each square foot of heating surface is capable of transmitting a certain amount of heat, depending upon the *conductivity of the material, the character of the surface, the temperature difference between the gas and the water, the location and arrangement of the tubes, the density of the gas, the velocity of the gas, and the time allowed for transmission of the heat.*

Thus one square foot of heating surface in the first pass of a water-tube boiler immediately over the incandescent mass of fuel may evaporate as high as 50 pounds of water per hour from and at 212 degrees F., whereas the same extent of surface close to the breeching evaporates less than one pound per hour. Because of this extreme variation it is convenient to assume a uniform heat transmission for the entire surface which will give the same total evaporation as that actually obtained. For maximum economy under *average* conditions of operation this gives a *mean* evaporation of 3 to 3.5 pounds of water per square foot per hour from and at 212 degrees F., which is equivalent to allowing 10 to 12 square feet per boiler horse power. By providing a large combustion chamber, increasing the extent of the first pass or the equivalent and by carrying a very thick bed of fuel a *mean* evaporation of 7 pounds per square foot per hour has been maintained with high economy. This corresponds to 5 square feet of heating surface per boiler horse power.

The maximum evaporation is limited only by the amount of coal which can be burned upon the grate. For example, a *mean* evaporation as high as 20 pounds* per square foot per hour has been effected in torpedo-boat practice, under intense forced draft, and 12 pounds per square foot per hour is not unusual in locomotive work. Such extreme, high rates of evaporation, however, are invariably obtained at the expense of fuel economy. In the very latest central stations the boiler and settings are proportioned to operate at 100 per cent above standard rating with high over-all efficiency and at 200 per cent above rating with only a small drop in efficiency, but such results are not obtainable in the ordinary everyday boiler and setting.

Builders of return tubular and vertical fire-tube boilers allow from 11 to 12 square feet of heating surface per horse power; water-tube boilers are rated at 10 square feet per horse power, and Scotch-marine boilers at 8 square feet per horse power.

See, also, paragraph 81, Effect of Capacity on Efficiency.

* Eng. Mag., Jan., 1912, p. 504.

The following table shows approximately the relation between boiler horse power and heating surface for different rates of evaporation:

EVAPORATION FROM AND AT 212 DEGREES F. PER SQUARE FOOT PER HOUR.

2	2.5	3.0	3.5	4	5	6	7	8	9	10
SQUARE FEET HEATING SURFACE REQUIRED PER HORSE POWER.										
17.3	13.8	11.5	9.8	8.6	6.8	5.8	4.9	4.3	3.8	3.5

Efficiency of Boiler Heating Surface: Trans. A.S.M.E., 18–328, 19–571. Kent, "Steam Boiler Economy" (John Wiley & Sons), Chapter IX.

The *Nature of True Boiler Efficiency:* Jour. West. Soc. Engrs., Sept. 18, 1907.

Heat Transference through Heating Surface: Engineering, 77–1.

77. The Horse Power of a Boiler. — A boiler horse power is equivalent to the evaporation of 34.5 pounds of water per hour from a temperature of 212 degrees F. to steam at atmospheric pressure. This corresponds to 33,479 B.t.u. per hour. Since the power from steam is developed in the engine and the boiler itself does no work, the above measure of capacity is merely conventional. Thus one boiler horse power will furnish sufficient steam to develop about three actual horse power in the best compound condensing engine, but only one-half horse power in a small non-condensing engine. Boilers should be purchased on the basis of heating surface and not on the horse-power rating, since one bidder may offer a boiler with, say, 5 square feet of heating surface per horse power and another with 10 square feet, both being capable of the required evaporation, but the one with the small heating surface (which will, of course, be the cheaper boiler) will have considerably less reserve capacity. Manufacturers ordinarily rate their boilers on the basis of from 10 to 12 square feet of heating surface per horse power, and the power assigned is called the *builder's rating.* As this practice is not uniform, bids and contracts should always specify the amount of heating surface to be furnished. According to the recommendations of the American Society of Mechanical Engineers, "A boiler rated at any stated capacity should develop that capacity when using the best coal ordinarily sold in the market where the boiler is located, when fired by an ordinary fireman, without forcing the fires, while exhibiting good economy. And, further, the boiler should develop at least one-third more than stated capacity when using the same fuel and operated by the same fireman, the full draft being employed and the fires being crowded; the available draft at the damper, unless otherwise understood, being not less than one-half-inch water column.

In determining the boiler horse power required for a given engine horse power it is convenient to estimate the steam consumption of the engine under actual conditions and then ascertain the equivalent evaporation from and at 212 degrees F. For example, assume a single non-condensing engine developing 20 horse power to use 50 pounds of steam per horse-power hour, or 1000 pounds steam per hour; steam pressure, 80 pounds per square inch; feed-water temperature 120 degrees F. Required the boiler horse power necessary to furnish this quantity of steam.

From equation (27), the factor of evaporation is

$$F = \frac{\lambda - q_2}{970.4} = \frac{1185.3 - 87.91}{970.4} = 1.131.$$

One thousand pounds of steam under the given conditions are therefore equivalent to $1000 \times 1.131 = 1131$ pounds from and at 212 degrees F.

The boiler horse power necessary to furnish steam for the 20-horse-power engine will be

$$\text{Boiler horse power} = \frac{1131}{34.5} = 32.8.$$

Example: A 15,000-kilowatt steam turbine and auxiliaries require 14.7 pounds of steam per kilowatt-hour at rated load; steam pressure 200 pounds per square-inch gauge; superheat 150 degrees F.; feed-water temperature, 179 degrees F.

Required the boiler horse power necessary to furnish this quantity of steam.

The heat furnished to the turbine and auxiliaries per kilowatt-hour is

$$\begin{aligned} w\{\lambda + C_p t_s - q_2\} &= 14.7\,\{1199.2 + 0.57 \times 150 - 146.88\} \\ &= 16{,}724 \text{ B.t.u.,} \end{aligned}$$

$$\text{Boiler horse power} = \frac{15{,}000 \times 16{,}724}{33{,}479} = 7500 \text{ (approx.)}.$$

For forced capacity of boilers, see Table 30.

78. Grate Surface. — The amount of fuel which can be burned per hour limits the amount of water evaporated per unit of time and depends upon the extent and nature of the grate surface, the character of the fuel and the draft. In locomotive and torpedo-boat practice space limitations necessitate the use of small grates and the rate of combustion is primarily a direct function of the draft. In stationary practice there is a wide permissible range in proportioning the grate surface since a given rate of combustion may be effected with large grate surface and light draft or with small grate surface and strong

draft. In a general sense the best results are obtained with a small grate and a high rate of combustion, but in the majority of installations the draft is comparatively feeble and a liberal grate area is necessary. So much depends upon the grade and size of the fuel that general rules for proportioning the grate surface are apt to lead to serious error. A liberal allowance of grate surface is desirable for hand-fired furnaces with natural draft, particularly if the ash is easily fusible, tending to choke the grate, but with forced draft and automatic stokers the best results are obtained with a thick fire and small grate surface. The relation between draft and rate of combustion for various sizes and kinds of coals is shown in Fig. 155.

A number of boiler tests made by Barrus ("Boiler Tests") showed that the best economy with anthracite coal, hand-fired, was obtained with an average ratio of grate surface to heating surface of 1 to 36, and at a rate of combustion of approximately 12 pounds of coal per square foot of grate surface per hour. In these tests a variation in grate and heating-surface ratio of 1 to 36 up to 1 to 46 gave practically no difference in economy. With bituminous coal the tests showed that an average ratio of 1 to 45 gave the best results and at a rate of combustion of 24 pounds of coal per square foot of grate surface per hour.

Tests made by Christie (Trans. A.S.M.E., 19–330) gave an average combustion of 13 pounds of anthracite per square foot of grate per hour for maximum efficiency and 24 pounds of bituminous.

Current central-station practice gives average rates of combustion as follows:

POUNDS OF COAL BURNED PER HOUR PER SQUARE FOOT OF GRATE SURFACE.

(Natural Draft.)

Anthracite, nut	15–20	Semi-bituminous, run of mine	20–30
Anthracite, pea	12–18	Semi-bituminous, screenings	20–30
Anthracite, buckwheat No. 1	8–12	Bituminous, run of mine	20–45
Anthracite, buckwheat No. 3	6–10	Bituminous, screened nut	20–40
Semi-anthracite, run of mine	18–25	Bituminous, screenings	20–35
Semi-anthracite, screenings	12–22	Bituminous, slack	18–30

With forced draft these rates of combustion may be greatly increased. Some idea of the extreme rate of combustion in modern locomotive practice may be obtained from the following figures which give the pounds of coal burned per hour per square foot of grate surface for various conditions of operation:

Maximum rate	200	Average rate	80
Very high rate	150	Economical rate	60
Average high rate	100	Low rate	50

Table 26 gives the relation between heating and grate surface in a number of recent boiler installations using different kinds of coal, and is illustrative of current practice.

In proportioning the grate surface for a proposed installation the principal factor considered is the character of the fuel, a study being made of the various fuels available, and the one selected which gives the highest evaporation per dollar (all items entering into the handling and combustion of the fuel being considered). This information may usually be obtained from records of plants using the same grade of fuel and grates similar to those intended for the proposed plant.

79. Boiler and Furnace Efficiency. — A perfect or ideal boiler and furnace is one which transmits to the water in the boiler the total heat of the fuel. In order to effect this result combustion must be complete, there must be no radiation or leakage losses and the products of combustion must be discharged at the initial temperature of the fuel. No commercial form of steam boiler can fulfill these conditions, hence the amount of heat absorbed by the boiler will always be less than the calorific value of the fuel.

The efficiency of the boiler and grate, and that of the boiler alone as recommended by the A.S.M.E., Rules for Conducting Boiler Tests, (Jour. A.S.M.E., Nov., 1912) may be expressed as

$$\text{Efficiency of boiler and grate} = \frac{\begin{array}{c}\text{Heat absorbed by the boiler}\\ \text{per pound of } \textit{coal as fired}\end{array}}{\begin{array}{c}\text{Calorific value of one pound}\\ \text{of } \textit{coal as fired}\end{array}}, \qquad (32)$$

and that of the boiler alone,

$$\text{Efficiency of boiler} = \frac{\begin{array}{c}\text{Heat absorbed by the boiler per pound}\\ \text{of } \textit{combustible burned} \text{ on the grate,}\end{array}}{\begin{array}{c}\text{Calorific value of one pound of } \textit{com-}\\ \textit{bustible as fired.}\end{array}} \qquad (32a)$$

The calculation of these efficiencies is illustrated by the following example:

DATA AS OBSERVED.

Steam pressure, pounds per square inch (gauge)	151.0
Barometer, inches of mercury	28.5
Temperature of feed water, degrees F.	161.8
Temperature of the furnace, degrees F.	2100.0
Temperature of flue gases, degrees F.	480.0
Temperature of boiler room, degrees F.	60.0
Quality of steam, per cent	98.0
Water apparently evaporated, pounds per hour	86,000
Coal as fired, pounds per hour	10,000
Refuse removed from ash pit, pounds per hour	1600

COAL ANALYSIS, PER CENT OF COAL AS FIRED.

Moisture	8
Ash	12

B.t.u. per pound, 11,250.

CALCULATED DATA.

Water apparently evaporated per pound of coal as fired, pounds = 86,000 ÷ 10,000 = 8.60.

Factor of evaporation* = [0.98 × 856.8 + 338.2 − (161.8 − 32)] ÷ 970.4 = 1.08.

Equivalent evaporation per pound of coal as fired, pounds = 8.6 × 1.08 = 9.288.

Heat absorbed by the boiler per pound of coal as fired, B.t.u. = 9.288 × 970.4 = 9,013.0.

Efficiency of boiler and grate, per cent = (9.013 ÷ 11,250) 100 = 80.11.

Refuse in ash referred to coal as fired, per cent = (1600 ÷ 10,000) 100 = 16.0.

Combustible burned on the grate, per cent of coal as fired = 100 − (8 + 16) = 76.0.

Equivalent evaporation per pound of combustible burned, pounds = 9.288 ÷ 0.76 = 12.221.

Heat absorbed per pound of combustible burned, B.t.u. = 12.221 × 970.4 = 11,860. Combustible as fired, per cent = 100 − (8 + 12) = 80.0.

Calorific value of the combustible as fired, B.t.u. = 11,250 ÷ 0.80 = 14,062.

Efficiency of the boiler, per cent = (11,860 ÷ 14,062) 100 = 84.34.

The efficiency of the grate alone might be expressed as

$$\text{Efficiency of grate} = \frac{\text{Efficiency of boiler and grate}}{\text{Efficiency of boiler}},$$

which is equivalent to

$$\text{Efficiency of grate} = \frac{\text{Combustible actually burned}}{\text{Combustible fired}}.$$

For the problem cited above,

$$\text{Efficiency of grate} = 100\left(\frac{80.11}{84.34}\right) = 95 \text{ per cent},$$

or,

$$\text{Efficiency of grate} = 100\frac{76}{80} = 95 \text{ per cent}.$$

This offers a good check on the calculations.

For oil fuel furnaces and coal furnaces equipped with stokers and forced draft appliances the *net* efficiency of the boiler and furnace may be taken as the boiler and furnace efficiency minus the equivalent heat required to feed the fuel and to create the draft.

Since the commercial form of boiler cannot possibly absorb all of the heat generated by the combustion of the fuel some authorities are of the opinion that the "true" efficiency of the boiler should be defined as the ratio of the heat absorbed to that actually *available*. Thus the U. S. Geological Survey defines the heat absorbed as the difference between the heat generated in the furnace and that discharged into

* See footnote, par. 74.

the flue, and the *available* heat is defined as the difference between the heat generated in the furnace and that discharged by the products of combustion at the temperature of the saturated steam.

If w = weight of the products of combustion, pounds per hour,
t_f = temperature of the furnace, degrees F.,
t_c = temperature of the flue gases, degrees F.,
t_s = temperature of the saturated steam, degrees F.,
t = temperature of the boiler room, degrees F.,
c_f, c_c, c_s = mean specific heat of the products of combustion for temperature ranges t to t_f, t_c, t_s respectively.

Then
wc_ft_f = heat generated in the furnace above t degrees F., B.t.u. per hour,
wc_ct_c = heat carried away by the flue gases above t degrees F.,
wc_st_s = heat carried away by the flue gases, if the temperature were lowered to t_s degrees F.,

$$E_1 = \text{the "true" boiler efficiency} = \frac{wc_ft_f - wc_ct_c}{wc_ft_f - wc_st_s},$$

$$E_1 = \frac{c_ft_f - c_ct_c}{c_ft_f - c_st_s}. \tag{33}$$

For most practical purposes it is sufficiently accurate to assume a constant value for the mean specific heats since the actual variation influences the result but slightly.

Assuming
$$c_f = c_c = c_s,$$
$$E_1 = \frac{t_f - t_c}{t_f - t_s}. \tag{34}$$

For the problem cited above

$$E_1 = 100\left(\frac{2100 - 480}{2100 - 366}\right) = 93.4 \text{ per cent.}$$

R. S. Hale (Trans. A.S.M.E., 20–769) gives as the efficiency of the furnace or combustion

$$\text{Efficiency of furnace} = \frac{S + F}{H}, \tag{35}$$

in which S = B.t.u. absorbed by the boiler per pound of dry coal,
F = B.t.u. lost in the flue gases per pound of dry coal,
H = calorific value of one pound of dry coal.

The efficiency of the ideal or perfect steam boiler may be expressed as

$$E_2 = \frac{H - I}{H}, \tag{36}$$

in which H = calorific value of the coal as fired,
I = inherent losses as analyzed in paragraph 30, B.t.u. per pound of coal as fired.

The efficiency ratio or the extent to which the theoretical possibilities are realized may be expressed as

$$E_3 = \frac{E}{E_2}, \tag{37}$$

in which

E = efficiency of the boiler and grate (A.S.M.E. code),
E_2 = as in equation (36).

The chief objection to the various efficiencies as defined in equations (34) to (36) is the difficulty of determining with any degree of accuracy the weight of the flue gases and the mean furnace temperature. For this reason commercial tests of boilers include only the efficiencies as recommended by the A.S.M.E. code.

Furnace Efficiency: Joseph Harrington, Jour. W. Soc. Engrs., Sept. 23, 1912.

TABLE 24.

RELATION BETWEEN FUEL CONSUMPTION AND BOILER AND FURNACE EFFICIENCY.

(Pounds of Fuel Burned per Boiler-Horse-Power Hour.)

Calorific Value of Fuel, B.t.u. per Pound.	Boiler and Furnace Efficiency.									
	40	45	50	55	60	65	70	75	80	85
7,500	11.17	9.91	8.94	8.12	7.45	6.87	6.37	5.95	5.58	5.25
8,000	10.45	9.30	8.37	7.60	6.97	6.43	5.98	5.58	5.22	4.92
8,500	9.84	8.75	7.87	7.12	6.56	6.05	5.62	5.25	4.97	4.63
9,000	9.30	8.25	7.45	6.76	6.20	5.72	5.31	4.96	4.65	4.36
9,500	8.80	7.83	7.05	6.40	5.87	5.41	5.02	4.69	4.40	4.14
10,000	8.37	7.44	6.70	6.09	5.58	5.15	4.79	4.46	4.18	3.94
10,500	7.98	7.09	6.39	5.80	5.86	4.90	4.56	4.26	3.99	3.76
11,000	7.60	6.79	6.09	5.52	5.06	4.67	4.34	4.05	3.80	3.59
11,500	7.28	6.49	5.83	5.29	4.85	4.47	4.16	3.88	3.64	3.45
12,000	6.97	6.22	5.58	5.06	4.65	4.28	3.99	3.72	3.48	3.28
12,500	6.69	5.97	5.35	4.86	4.46	4.11	3.82	3.57	3.34	3.14
13,000	6.44	5.74	5.15	4.68	4.29	3.96	3.68	3.43	3.22	3.02
13,500	6.20	5.52	4.96	4.51	4.18	3.81	3.54	3.31	3.10	2.91
14,000	5.98	5.33	4.79	4.35	3.99	3.68	3.42	3.19	2.99	2.81
14,500	5.77	5.15	4.62	4.20	3.84	3.54	3.30	3.08	2.88	2.72
15,000	5.58	4.96	4.47	4.06	3.72	3.43	3.19	2.98	2.79	2.64

80. Boiler Performances. — Table 26 is compiled from a number of tests of different types of boilers with various types of grates and characters of fuel. Although some of the tests show a combined efficiency of boiler and grate as high as 83 per cent, such a performance cannot be expected for continuous operation under the average conditions of practice. In pumping stations or in plants where there

are no peak loads and the boiler may be operated under a practically constant set of conditions a continuous efficiency of 75 per cent has been realized with coal as fuel and 80 per cent with crude oil, though these figures are exceptional. In very large central stations, with the usual peak loads in the morning and evening and long banking periods, an average efficiency throughout the year of 65 per cent is possible, though a good figure is not far from 60 per cent. In large isolated stations with variable loads good practice gives an average of 60 per cent. Small stations though showing an efficiency as high as 75 per cent at times seldom average 50 per cent for the year. The usual discrepancy between efficiency as determined by special tests and

TABLE 25.

RELATION BETWEEN RATE OF EVAPORATION PER POUND OF FUEL AND BOILER AND FURNACE EFFICIENCY.

Pounds of Water Evaporated per Hour from and at 212 Deg. F. per pound of Fuel.

Calorific Value of Fuel, B.T.U. per Pound.	Boiler and Furnace Efficiency.									
	40	45	50	55	60	65	70	75	80	85
7,500	3.09	3.48	3.86	4.25	4.64	5.02	5.41	5.80	6.18	6.57
8,000	3.30	3.71	4.12	4.55	4.95	5.36	5.77	6.18	6.60	7.01
8,500	3.51	3.94	4.38	4.81	5.26	5.70	6.14	6.57	7.01	7.45
9,000	3.71	4.18	4.64	5.10	5.56	6.04	6.50	6.96	7.42	7.90
9,500	3.92	4.41	4.90	5.39	5.88	6.47	6.86	7.35	7.85	8.33
10,000	4.12	4.64	5.16	5.66	6.19	6.70	7.21	7.74	8.25	8.76
10,500	4.31	4.86	5.40	5.94	6.48	7.01	7.55	8.10	8.64	9.17
11,000	4.52	5.09	5.65	6.22	6.79	7.35	7.91	8.48	9.05	9.61
11,500	4.74	5.31	5.91	6.50	7.10	7.69	8.28	8.86	9.45	10.0
12,000	4.94	5.55	6.16	6.78	7.40	8.01	8.64	9.25	9.86	10.5
12,500	5.14	5.78	6.42	7.06	7.70	8.35	9.00	9.64	10.3	11.0
13,000	5.35	6.01	6.69	7.35	8.01	8.69	9.35	10.0	10.7	11.4
13,500	5.56	6.25	6.95	7.65	8.34	9.03	9.72	10.4	11.1	11.8
14,000	5.75	6.48	7.20	7.91	8.64	9.35	10.1	10.8	11.6	12.2
14,500	5.96	6.70	7.45	8.20	8.95	9.70	10.5	11.2	12.0	12.7
15,000	6.18	6.95	7.72	8.50	9.26	10.1	11.8	11.6	12.4	13.1

every-day operation is due to the fact that the efficiency test is usually conducted under ideal conditions: the boiler surfaces are cleaned, the rate of combustion carefully adjusted for maximum economy, and special attention given to the firing, whereas in actual practice these refinements are seldom attempted. Much depends upon the efficiency of the boiler-room staff, the character of furnace and fuel, draft, and the load factor. From the commercial standpoint the performance is conveniently expressed in terms of the "cost to evaporate 1000 pounds

TABLE 26.

EXAMPLES OF CURRENT STEAM BOILER PERFORMANCES.

Authority.	Type of Boiler.	Builder's Rating, Horse Power.	Percentage of Builder's Rating Developed.	Method of Firing.	Kind of Coal.	Grate Surface, Sq. Ft.	Ratio Heating Surface to Grate Area.	Coal Burned per Sq. Ft. Grate per Hr.	Apparent Evaporation per Lb. of Coal as Fired.	Equivalent Evaporation per Lb. of Combustible.	Equivalent Evaporation per Sq. Ft. H. S. per Hr.	Combined Efficiency of Boiler and Grate.
1	Almy	351	129	Hand fired	Anthracite egg	86.0	40.9	18.7	8.04	10.78	4.49	67.6
2	Babcock & Wilcox	107	130	....do	Pocahontas lump	25.0	42.8	20.1	10.48	12.33	4.31	73.8
3	Do	231	173	....do	No. 3 buckwheat	......	59.5	31.8	9.25	10.29	5.98	71.0
3	Do	......	109	....do	....do	......	35.9	16.1	8.37	11.99	3.74	68.8
4	Do	250	86	....do	Anthracite pea	56.0	41.5	9.8	10.40	13.20	2.60	......
5	Do	412	105	Roney stoker	New River	74.8	55.0	21.0	9.68	12.40	3.60	74.9
6	Do	420	140	Bennis stoker	......	50.0	84.0	35.2	9.95	......	4.83	82.2
7	Do	500	220	Chain grate	No. 4 bituminous nut	90.0	55.6	43.2	7.80	11.80	7.59	72.0
8	Do	650	128	Taylor stoker	......	106.0	61.4	23.1	10.00	12.94	4.52	79.8
8	Do	650	185	....do	......	106.0	61.4	35.1	9.45	12.21	6.51	72.7
9	Berry	250	123	Hand fired	Bituminous	46.0	40.7	20.3	9.60	11.40	5.68	......
10	Bigelow-Hornsby	169	145	......	Nixon's navigation	40.0	42.0	18.6	9.58	12.18	4.57	73.4
11	Edge Moor	600	188	Taylor stoker	Clearfield, mine run	92.0	66.5	42.5	8.14	11.60	6.36	71.3
12	Climax vertical	500	......	Hand fired	Nixon's navigation	64.0	78.0	17.6	9.10	......	......	79.3
12	Do	1000	......	Wilkinson stoker	George's Creek	113.6	73.5	27.0	8.96	......	......	72.7
13	Heine	210	98	Hand fired	Bituminous, mine run	40.5	50.0	18.4	7.92	10.74	3.53	66.8
14	Do	350	109	Chain grate	Bituminous screenings	72.0	48.5	23.9	6.85	9.64	4.18	66.8
15	Italian torpedo	111.5	583	......	......	28.0	39.8	121.0	6.00	......	20.09	......
16	Keeler	250	100	Hand fired	No. 2 washed bituminous	48.0	52.0	24.0	6.68	8.28	3.50	......
17	Manning	150	147	Taylor stoker	Pocahontas, mine run	29.0	56.5	29.0	9.86	12.24	4.89	76.4
18	Do	150	129	Hand fired	Pocahontas	28.7	48.2	12.3	8.22	12.00	4.90	66.9
19	Return tubular	80	218	Hawley	Ohio bituminous, mine run	22.0	43.6	26.8	8.65	10.84	6.26	......
20	Do	100	108	Chain grate	Semi-bituminous	20.0	50.0	19.2	8.92	11.40	3.10	76.0
20	Do	100	273	....do	....do	20.0	50.0	55.0	7.67	9.68	7.86	64.0
21	Do	150	124	Hand fired	....do	39.0	46.0	20.1	7.80	9.00	4.48	60.0
21	Do	150	135	Chain grate	....do	35.0	51.5	18.0	10.56	12.00	4.81	80.0

22	Do	200	122	Hand fired	....do	39.0	59.0	20.5	9.35	11.67	3.60	70.2
23	Robb-Mumford	107	90	Rocking, hand fired	George's Creek	24.5	44.0	13.2	8.75	10.60	2.93	69.0
24	Rust water tube	335	105	Roney stoker	Bituminous, crushed lump	68.0	49.0	17.0	....	12.21	3.63	75.5
24	Do	335	210	....do	....do	68.0	49.0	38.0	....	10.86	7.26	68.9
25	Sederholm	300	......	Shaking grate	Illinois screenings	51.0	57.8	24.9	8.05	10.05	3.87	....
26	Scotch-marine	75	108	Hand fired	Bituminous, mine run	15.0	35.2	20.5	8.20	10.25	5.01	70.5
26	Do	250	89	....do	Anthracite chestnut	48.0	38.0	15.2	9.98	12.27	4.80	76.2
26	Stirling	350	98	Chain grate	No. 3 washed bituminous	82.5	42.5	17.0	8.20	11.70	3.40	70.2
27	Do	150	122	....do	New River	36.0	41.5	14.7	11.20	12.50	4.20	79.0
28	Do	542	136	Jones stoker	Bituminous nut and slack	81.2	66.7	25.6	10.68	12.92	4.70	83.0
28	Do	542	227	....do	....do	81.2	66.7	44.7	10.10	11.98	7.82	79.3
29	Do	2365	105	Roney stoker	Bituminous	446.0	53.0	16.7	9.75	10.52	3.63	77.8
29	Do	2365	197	....do	....do	446.0	53.0	33.6	8.86	10.66	6.75	75.6
29	Do	2365	107	Taylor stoker	....do	405.0	58.5	18.8	9.80	11.55	3.72	80.3
30	Wickes horizontal	150	100	Shaking grate	Bituminous, mine run	27.0	64.4	21.2	7.25	9.79	2.93	67.5
30	Do	200	99	Murphy stoker	Bellmore pea	42.0	54.3	20.0	8.42	11.27	3.00	75.4
30	Wickes vertical	225	107	....do	New River, mine run	39.0	68.6	18.0	9.46	12.68	3.11	79.5
26	Vertical tubular	40	95	Hand fired	Pocahontas	9.5	52.2	13.7	8.36	10.20	2.80	63.5
26	Do	60	104	....do	No. 3 washed bituminous	12.0	59.7	21.3	7.66	9.88	3.00	61.7

1. G. H. Barrus. 2. Eng. Rec., July 25, 1903, p. 102. 3. B. & W. "Steam." 4. Eng. Rec., March 9, 1901, p. 220. 5. Power, Dec., 1901, p. 26. 6. Eng. Rec., April 8, 1905, p. 404. 7. Commonwealth Edison Co. 8. N. Y. Edison, Waterside. 9. Catalogue, Berry Boiler Co. 10. Circular, Bigelow Co. 11. Power, March 21, 1911, p. 447. 12. Catalogue, Climax Boiler Co. 13. U. S. Geological Survey, Professional paper No. 48, Part II. 14. Bul. No. 22, Vol. 2, University of Illinois. 15. Eng. Mag., Jan., 1912, p. 504. 16. Chicago Stock Exchange Bldg. 17. Everett Mills, Lawrence, Mass. 18. J. E. Denton. 19. Circular, Hawley Down Draft Co. 20. N. Y. Central H. R. R. R. 21. Circular, Universal Chain Grate Co. 22. Power, Aug. 2, 1910. 23. Eng. Mag., April, 1904. 24. Eng. U. S., Feb. 15, 1908, p. 232. 25. Eng. Rec., Dec. 20, 1902, p. 584. 26. Tests conducted by the author. 27. Circular, Ironworks Co., New Jersey. 28. J. W. Hill. 29. Jour. A. S. M. E., Nov., 1911. 30. Catalogue, Wilkes Boiler Co.

of water from and at 212," or the "pounds of water evaporated per $1 of coal." Table 28 gives the results of a number of tests, made at the Armour Glue Works, Chicago, Ill., showing the cost of evaporating water with different grades of Illinois coal. The results were obtained from hand-fired Stirling boilers.

81. Effect of Capacity on Efficiency. — In general, as the horse power of a boiler increases above normal capacity the over-all efficiency will decrease, due to the fact that the furnace and gas passages are *ordinarily* proportioned to effect an evaporation of about 3.5 pounds of water from and at 212 degrees F. per square foot of heating surface per hour at rated load, the temperature of the escaping gases being from

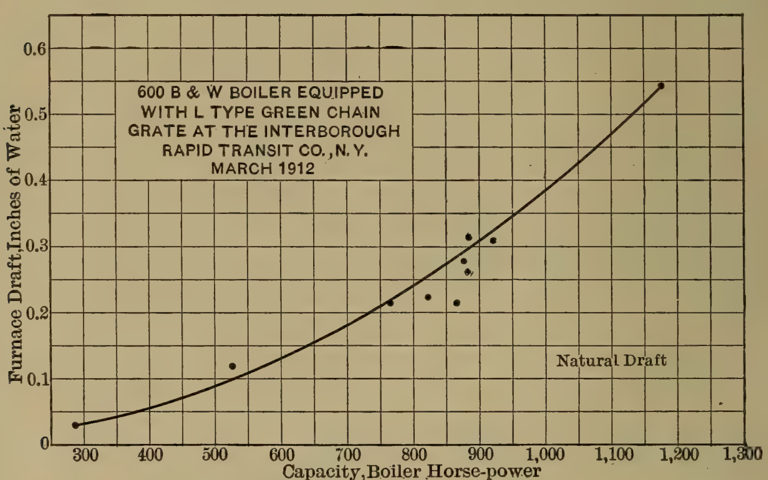


FIG. 56. Influence of Draft on Capacity.

150 to 200 degrees above that of the steam. To increase the rate of evaporation more coal must be burned per unit of time and consequently a larger volume of gas is generated. The larger the volume of gas the higher will be its velocity, which *finally* reaches a point where heating surface is insufficient in extent to absorb the extra heat and as a consequence the flue gas escapes at a higher temperature, resulting in lower boiler and furnace efficiency. See Fig. 57. With properly proportioned grate, furnace and gas passages a boiler may be operated at 100 per cent above standard rating with little or no decrease in overall efficiency. Figs. 56 to 62 illustrate various phases of boiler performances under different conditions of operation. These curves are of value simply as illustrations of the behavior in specific cases, and are not applicable to all types of boilers. See also Tables 30 and 32.

TABLE 27.

PRINCIPAL DATA AND RESULTS OF TESTS ON BOILER NO. 6, UNIT NO. 10, FISK ST. STATION. COMMONWEALTH EDISON CO., CHICAGO.

(B. & W. Boiler, " Standard " Setting.)

Water-heating Surface, 5000 Sq. Ft. Superheating Surface, 914 Sq. Ft. Chain Grate Surface, 90 Sq. Ft.

Test No.	Date, 1908.	Horse Power.	Eff'y, Per Cent.	H.P. per Sq. Ft. Grate.	Heat Lost in Refuse, Per Cent.	Total Heating Surface per H.P.	Super-heat of Steam, Deg. F.	Dry Coal per Sq. Ft. G. S. per Hour.
2	Mar. 9	873	67.4	9.70	2.8	6.76	197	41.2
4	" 10	873	69.0	9.52	2.8	6.89	195	39.1
6	" 11	852	67.3	9.47	2.8	6.93	189	38.9
8	" 16	836	65.3	9.29	6.4	7.06	174	39.5
10	" 17	870	68.8	9.67	5.0	6.78	180	39.3
14	" 19	920	66.2	10.22	9.2	6.42	187	43.7
16	" 23	900	69.5	10.00	4.0	6.56	181	40.5
18	" 24	916	69.1	10.18	5.5	6.44	190	41.6
20	" 26	912	69.2	10.13	4.4	6.48	179	41.2
22	" 27	906	67.7	10.07	4.1	6.52	194	42.5
24	" 30	925	69.8	10.28	2.8	6.38	179	41.6
26	" 31	894	69.4	9.93	5.2	6.60	170	40.6
28	Apr. 1	922	71.2	10.24	3.6	6.40	169	40.4
30	" 2	923	71.5	10.26	4.6	6.40	173	40.5
32	" 7	914	70.0	10.20	4.5	6.46	175	40.9
34	" 8	939	73.8	10.4	3.8	6.28	181	40.4
36	" 10	911	70.9	10.1	3.0	6.48	185	40.2
38	" 11	967	70.1	10.7	3.0	6.11	192	42.6
40	" 13	995	67.8	11.1	3.4	5.93	211	43.6
42	" 14	887	66.8	9.9	4.5	6.65	202	40.8
44	" 27	880	69.5	9.8	5.5	6.72	169	39.7
48	" 29	927	71.5	10.3	3.3	6.37	171	40.8
50	" 30	899	70.3	10.0	4.2	6.57	171	39.6
52	May 6	886	69.4	9.8	5.3	6.67	171	38.2
54	" 7	900	69.1	10.0	4.8	6.56	171	39.2
56	" 8	967	71.9	10.7	4.8	6.10	164	40.1
58	" 11	902	70.5	10.0	3.3	6.55	163	39.6
60	" 13	875	70.7	9.7	3.8	6.74	147	38.3
64	" 14	1102	72.0	12.2	4.8	5.35	180	43.2

TABLE 27.

PRINCIPAL DATA AND RESULTS OF TESTS ON BOILER NO. 6, UNIT NO. 10, FISK ST. STATION. COMMONWEALTH EDISON CO., CHICAGO.

(B. & W. Boiler, "Standard" Setting.)

Water-heating Surface, 5000 Sq. Ft. Superheating Surface, 914 Sq. Ft. Chain Grate Surface, 90 Sq. Ft.

Draft		B.T.U. per Pound Dry Coal.	Ash in Dry Coal, Per Cent.	Ash in Refuse, Per Cent.	Uptake Temp. Deg. F.	CO_2, Per Cent.	Heat Lost up Stack (Dry Gas), Per Cent.
Over Fire.	In Uptake.						
.87	1.34	11,634	18.46	82.33	466	6.9	....
.78	1.25	11,759	16.81	81.36	461	6.7	....
.83	1.25	12,039	16.08	80.03	463	7.7	15.6
.94	1.34	11,993	15.91	67.42	477	7.6	16.8
.84	1.24	11,909	15.71	71.32	475	7.9	16.2
.99	1.41	11,768	16.04	63.78	479	8.5	15.4
.77	1.17	11,846	16.68	79.04	483	9.1	14.0
.81	1.25	11,800	16.39	71.98	484	8.3	15.8
.77	1.21	11,846	15.51	78.53	486	9.0	14.5
.78	1.22	11,659	17.59	80.58	494	9.2	14.6
.68	1.28	11,800	16.22	82.97	487	8.8	15.1
.70	1.24	11,752	16.18	76.84	484	8.8	15.1
.62	1.21	11,862	15.38	82.99	480	9.2	14.1
.58	1.40	11,800	16.02	78.37	480	9.1	14.4
.73	1.24	11,815	16.84	77.84	494	9.0	14.7
.72	1.25	11,659	18.06	82.27	504	8.9	15.3
.65	1.13	11,831	17.15	86.92	493	9.7	13.4
.70	1.24	12,002	16.05	84.39	502	9.0	15.1
.71	1.23	12,469	14.87	82.14	522	9.7	13.3
.63	1.09	12,049	15.17	78.12	500	9.5	13.3
.71	1.26	11,801	15.75	77.21	470	8.3	15.7
.68	1.23	11,769	18.59	84.04	472	8.7	14.2
.66	1.27	11,955	16.11	79.30	473	7.9	16.1
.62	1.20	12,360	13.63	74.59	476	8.8	14.5
.66	1.31	12,298	13.62	75.19	480	9.0	14.4
.66	1.29	12,423	13.37	75.61	474	9.4	13.3
.92	1.18	11,956	17.45	83.24	451	9.2	12.5
.76	0.98	11,971	17.45	80.99	443	10.0	11.2
.68	1.15	13,126	10.24	70.90	487	10.4	12.1

TABLE 28.

RESULTS OF COAL TESTS AT ARMOUR GLUE WORKS, CHICAGO, AUG. 17, 1905.

Date of Test.	Name and Kind of Coal.	Railroad Car Number.	Cost per Ton Delivered.	Cost to Evaporate 1000 Pounds of Water.	Pounds Water Evaporated per $1.00 of Coal.
March 5, 1905..	Williamson County Coal Co.'s, mine run	C. C. C. & St. L. No. 26368	$1.90	$0.1531	6,532
March 3, 1905..	Harden & Hafer, mine run	S. I. No. 5735	1.70	0.1231	8,123
June 14, 1905 ..	Crerar-Clinch & Co., 2″ screenings.	I. C. No. 88362	1.50	0.1293	7,734
June 15, 1905 ..	....do................	I. C. No. 88362	1.50	0.1218	8,210
June 16, 1905 ..	....do................	I. C. No. 88362	1.50	0.1175	8,511
June 17, 1905 ..	Brackett Coal and Coke Co., lump.	C. & E. I., No. 8891.	1.65	0.122	8,197
June 19, 1905 ..	....do................	C. & E. I. No. 5002	1.65	0.1212	8,251
June 20, 1905 ..	....do................	C. & E. I. No. 5002	1.65	0.1352	7,396
July 1, 1905....	Kellyville Coal Co., mine run.	C. & E. I. No. 10030.	1.595	0.1355	7,380
July 6, 1905....	Brackett C. & C. Co., Keeler mine run.	C. & E. I. No. 12367	1.65	0.1236	8,091
July 28, 1905...	Kellyville Coal Co., washed pea.	C. & E. I. No. 6211.	1.50	0.1285	7,782
July 29, 1905...	....do................	C. & E. I. No. 6211	1.50	0.119	8,403
Aug. 5, 1905 ...	Dering Coal Co., mine run.	C. & E. I. No. 25125	1.575	0.125	8,000
Aug. 7, 1905 ...	Dering Coal Co., Sullivan Co., screenings.	E. & T. H. No. 5132.	1.40	0.11	9,091
Aug. 8, 1905 ...	Consolidated Indiana Coal Co., Sullivan Co., screenings.	E. & T. H. No. 3239	1.35	0.105	9,524
Aug. 9, 1905 ...	Screenings...........	E. & T. H. No. 6534	1.30	0.0973	10,277
Aug. 11, 1905 ..	Ziegler, screenings...	I. C. No. 81184	1.50	0.1047	9,551

TABLE 29.

RELATION BETWEEN CAPACITY AND EFFICIENCY.

(Evaporation from and at 212° F. per Square Foot of Heating Surface per Hour.)

2	2.5	3	3.5	4	5	6	8	10	12
Probable Relative Economy, Ordinary Installation.									
100	100	100	95	90	85	80	70	60	50
Probable Relative Economy, Latest Improved Installation.									
95	98	100	100	100	99	98	95	90	85

TABLE 30.

FLUE GAS TEMPERATURES CORRESPONDING TO FORCED CAPACITY OF BOILERS IN MODERN POWER PLANT INSTALLATIONS.

Plant.	Type of Boiler.	Rated Horse Power per Unit.	Heat Surface per Horse Power Developed.	Flue Temperature.	Builders' Rating, Per Cent.
Cambridge Steel Co.	B. & W.	400	5.14	485	194.5
Commonwealth Edison Co.	B. & W.	650	4.97	588	201.0
Detroit Edison Co.	Stirling	2365	4.75	651	211.3
Everett Mills	Manning	130	6.0	599	150.0
Narragansett Electric Lighting Co.	B. & W.	440	5.5	544.2	180.4
National Museum	Geary, W. T.	182.8	6.4	430	155
N. Y. Central R.R., West Albany	Edgemoor	600	5.28	543	193
N. Y. Central & H. R. R.R.	Ret. Tub.	100	4.4	630	273
N. Y. Edison Waterside	B. & W.	650	5.48	550	179
Old Colony St. Ry.	B. & W.	687.5	5.25	599	190
Union Gas & Electric Co.	Stirling	542	4.43	622	227

TABLE 31.

PRINCIPAL DATA AND RESULTS OF TESTS ON 2365-RATED-HORSE-POWER STIRLING BOILERS AT THE DELRAY STATION OF THE DETROIT EDISON COMPANY.

Tests with Roney Stoker. Résumé of Principal Results.

No. of Test.	Length, Hr.	Per Cent Rating.	B.t.u. in Coal.	Per Cent Ash in Dry Coal.	Efficiency.	Per Cent Steam used by Stoker Engines and Steam Jets.	Per Cent Combustible in Ash.	Temp. of Flue Gases Leaving Boiler, Deg. Fahr.
1	25	105.0	14,362	5.98	77.84	....	19.6	576
2	24	80.0	14,225	6.52	79.88	....	17.9	480
3	24	113.8	14,308	7.40	77.45	0.63	24.4	542
4	30	152.4	13,756	6.54	75.78	1.58	30.8	670
5	24	94.0	13,896	6.89	81.15	1.75	31.6	483
6	24	150.7	14,037	6.13	75.28	1.45	26.7	662
16	32	98.6	14,476	9.68	80.98	1.34	34.1	460
17	16.5	193.3	14,493	8.24	76.73	1.39	24.6	636
18	24	195.7	13,689	9.81	75.57	1.32	23.2	694
2–4†	90	119.8	14,098	6.81	76.13	....	25.8	572
5–6†	55	127.3	13,977	6.84	76.23	....	29.4	575

† Including periods between tests.

Tests with Taylor Stoker. Résumé of Principal Results.

No. of Test.	Length, Hr.	Per Cent Rating.	B.t.u. in Coal.	Per Cent Ash in Dry Coal.	Efficiency.	Per Cent Steam used by Stoker Auxiliaries.*	Per Cent Combustible in Ash.	Temp. of Flue Gases Leaving Boiler, Deg. Fahr.
7	24	151.2	14,000	7.03	77.07	2.61	31.5	575
8	24	107.9	13,965	6.34	80.28	2.44	27.1	493
9	50	162.8	13,998	6.75	77.85	2.87	31.3	574
10	48	92.9	14,188	9.90	77.90	2.63	27.2	487
11	26.5	211.3	14,061	9.55	75.84	3.41	36.1	651
12	48	121.3	14,010	8.09	79.24	2.57	27.6	535
14	24	185.2	14,272	8.71	76.42	2.95	28.8	647
15†	24	123.1	14,213	8.34	74.90	2.77	30.1	561
7–9‡	109	140.0	13,983	7.22	77.66	2.68	29.9	545
10–11‡	80.5	132.8	14,095	9.58	75.66	3.04	31.1	542

* Engines driving stokers and steam-turbine driving fan.

† In test No. 15 the fires were banked for 7½ hours and the averages include this period.

‡ Including periods between tests.

In nearly all stations the boilers must have sufficient overload capacity to take care of peak loads or to allow some of the boilers to be shut down for cleaning or repairs, since the installation of sufficient rated boiler capacity would be expensive and in many instances prohibitive in cost. In small stations, however, too large a boiler capacity frequently is to be preferred to an overloaded installation, since the extra first cost of the former may be less than the loss due to poor efficiency and depreciation in the latter.

As far as forcing is concerned the fire-tube boiler is as effective as the water-tube, more depending upon the furnace, grate surface, draft and character of fuel than upon the type of boiler. All boilers are subject to more or less priming at heavy overloads, and the overload capacity is often limited on this account.

The Forcing Capacity of Fire-Tube Boilers: F. W. Dean, Trans. A.S.M.E., 26–92.
Increasing Capacity of Steam Boilers: Kreisinger and Ray, Power, May 24, 1910.

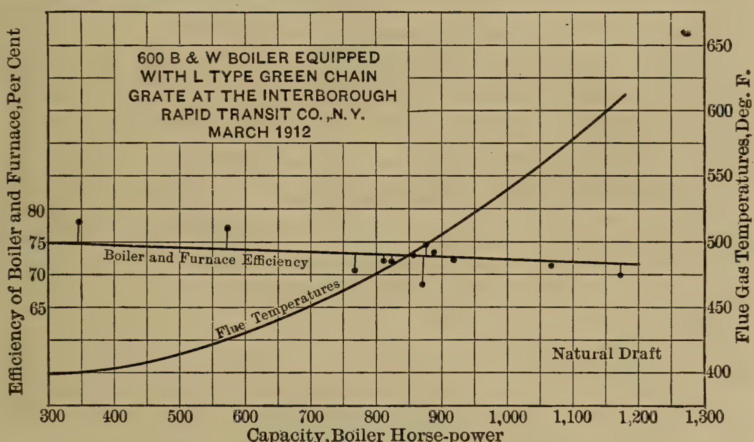


FIG. 57. Effect of Rate of Driving on Capacity.

82. Thickness of Fire. — For a given furnace and boiler, quality and size of fuel and intensity of draft, a certain depth of fuel will give maximum efficiency. Too thin a fire results in an excess of air and too thick a fire in a deficiency, the economy being lowered in either case. On account of the number of conditions upon which the proper thickness depends, it can only be determined for a particular case by actual test, the available data being insufficient for drawing conclusions. The curves in Fig. 63 are plotted from a series of tests made on a 350-horsepower Stirling boiler equipped with chain grate at the power plant of the Armour Institute of Technology. The damper was left wide open throughout the test and the speed of the grate kept constant. Ratio of grate to heating surface, 1 to 42. Carterville washed coal No. 4 was

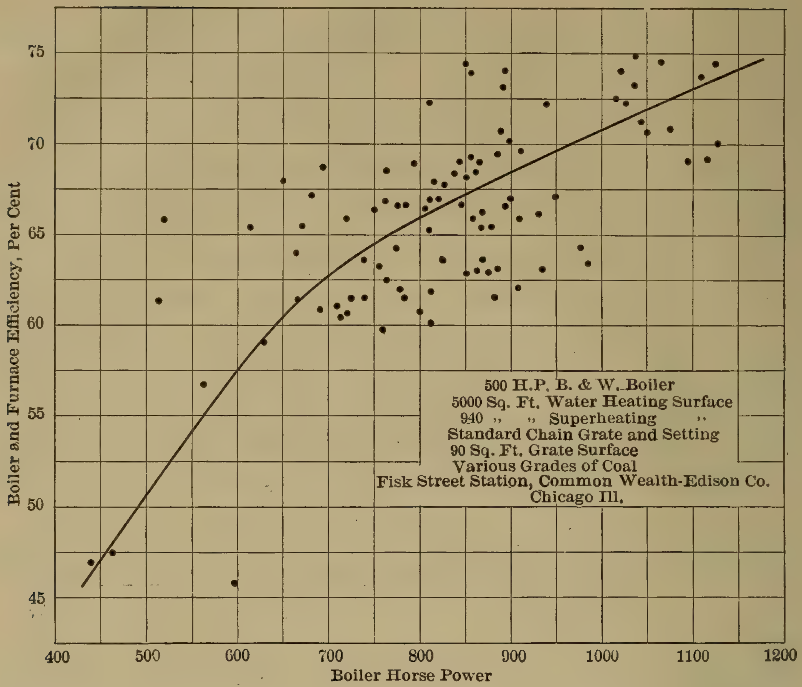


FIG. 58. Relation Between Efficiency and Capacity; 500 H.P. Boiler, Fisk Street Station, Commonwealth Edison Co., Chicago.

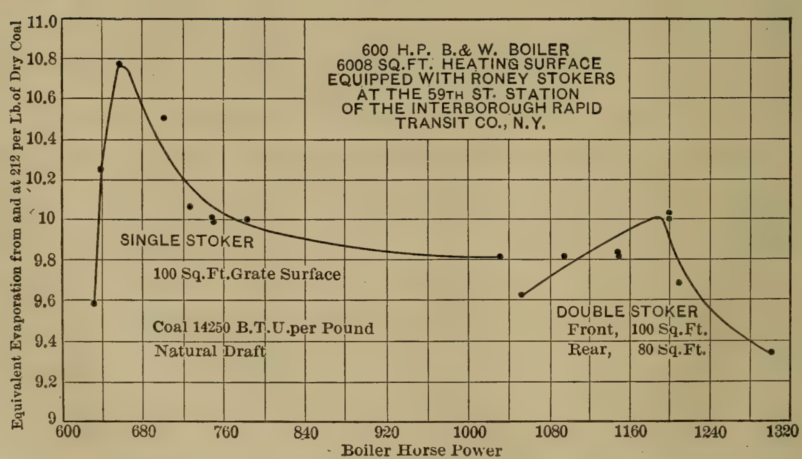


FIG. 59. Effect of Rate of Driving on the Efficiency of a 600 H.P. B. & W. Boiler.

used in all tests. The curves in Fig. 64 refer to the performance of a 150-horse-power water-tube boiler equipped with chain grate at the University of Illinois Engineering Experiment Station at Urbana.

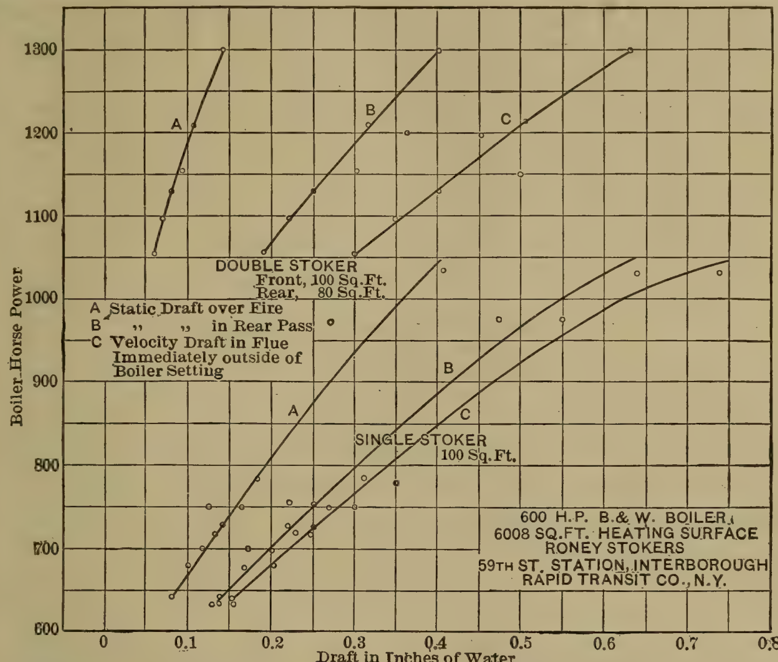


FIG. 60. Influence of Draft on the Capacity of a 600-Horse-power B. & W. Boiler.

The curves in Fig. 65 are plotted from a series of tests on a 500-horse-power Babcock and Wilcox boiler equipped with chain grate at the Fisk

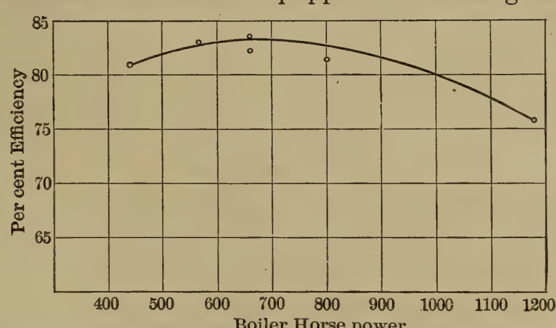


FIG. 61. Relation between Efficiency and Capacity — Oil Fuel.

Street station of the Commonwealth Edison Company, Chicago, Ill. In these tests the conditions of operation are not exactly comparable, but they serve to show the variation of economy with thickness of fire

in each case. In general, with natural draft, fine sizes of coal necessitate thin fires, since they pack so closely as to greatly restrict the draft. Thin fires require closer attention to prevent holes being burned in spots, and respond less readily to sudden demands for steam, but have

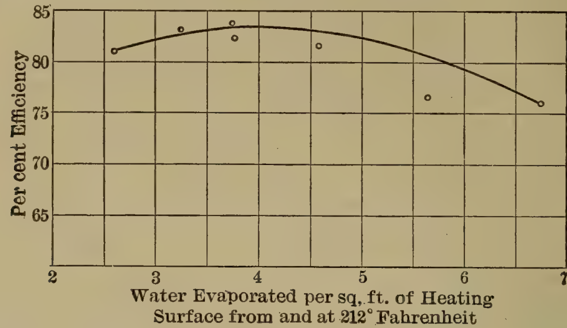


FIG. 62. Relation between Efficiency and Evaporation — Oil Fuel.

the advantage of letting the air required pass through the grate, whereas thick fires often require air to be supplied above the grate to insure complete combustion. Thick fires require less attention and hence are preferred by firemen. Where sufficient draft is available thick fires

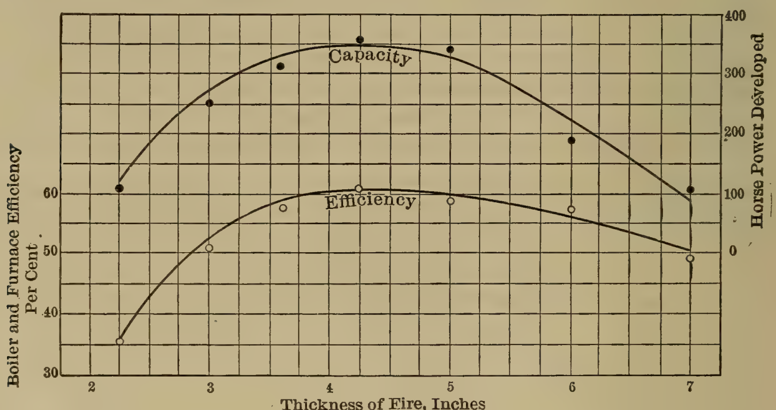


FIG. 63. Effect of Thickness of Fire on the Capacity and Efficiency of a 350-Horsepower Stirling Boiler, Equipped with Chain Grate.

are more efficient than thin ones, as the air excess is more readily controlled.

83. Influence of Initial Temperature on Efficiency. — In general the higher the initial temperature of the furnace the greater will be the efficiency of the *heating surface,* since the heat transmitted varies almost directly with the difference of temperature between the water and the products of combustion. If the heating surface is properly distributed

so that the final temperature of the escaping gas remains constant, the efficiency of the boiler and furnace will increase as the initial temperature increases, though not in direct proportion. This is on the assumption that the amount of heat generated per hour is the same throughout all ranges in temperatures. With a condition where the amount of heat generated remains constant and the initial temperature varies, the final temperature of the escaping gases remains practically constant,

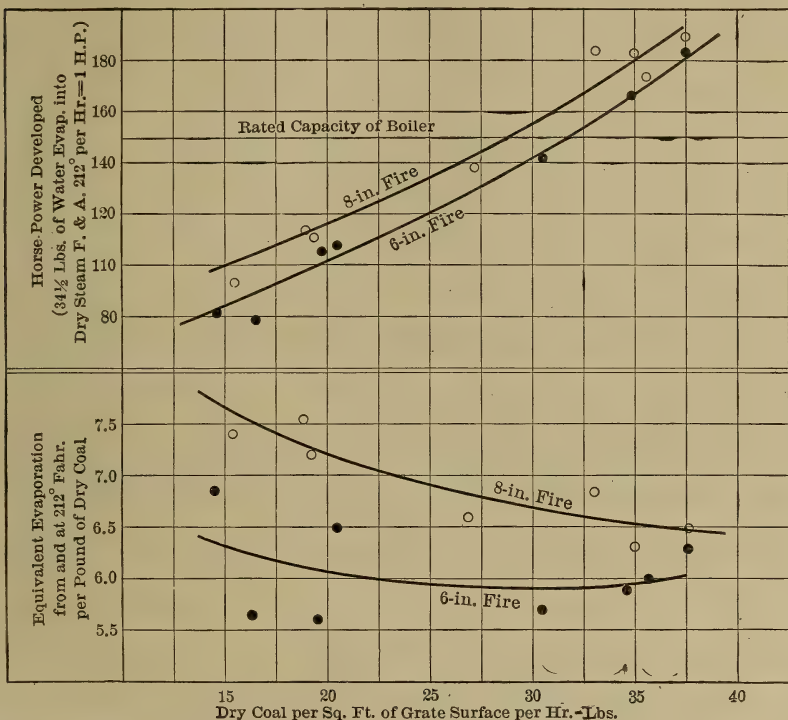


FIG. 64. Effect of Thickness of Fire on the Capacity and Efficiency of a 150-Horsepower Water-tube Boiler.

and in such cases high initial temperatures are productive of high *boiler* and *furnace efficiencies.* In practice these conditions are seldom realized and high furnace temperatures are not necessarily productive of high boiler and furnace efficiencies. Some tests show a decided gain in efficiency with the higher furnace temperatures ("Some Performances of Boilers and Chain-grate Stokers, with Suggestions for Improvements," A. Bement, Jour. West. Soc. Engrs., February, 1904), and others show little if any improvement ("A Review of the United States Geological Survey Fuel Tests under Steam Boilers," L. P. Brecken-

ridge, Jour. West. Soc. Engrs., June, 1907). The majority of high-efficiency records, however, are associated with high furnace temperatures.

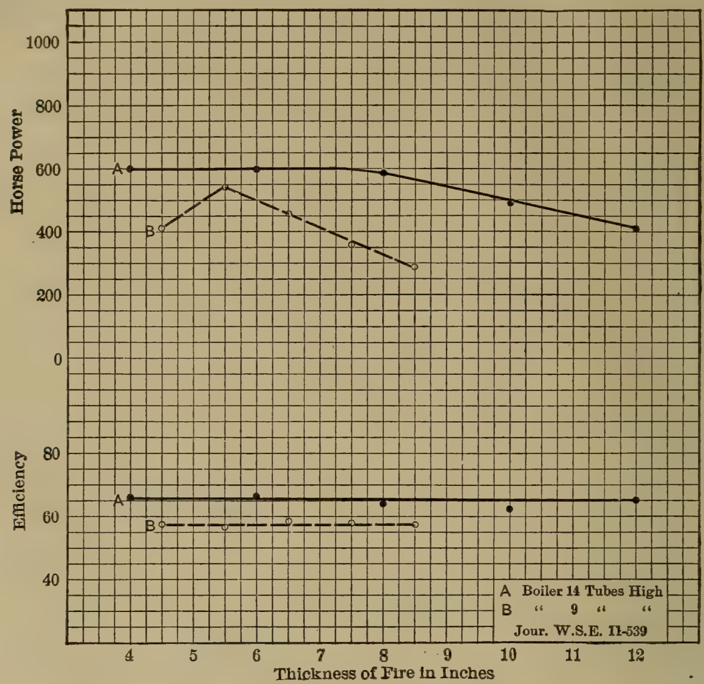


FIG. 65. Effect of Thickness of Fire on the Capacity and Efficiency of a 500-Horse-power Babcock and Wilcox Boiler.

TABLE 32.

TEMPERATURE DROP OF GASES THROUGHOUT BOILER.

(650 H.P., B. & W. Boiler, Waterside Station of N. Y. Edison Co.)

Per Cent of Rating.	Boiler and Grate Efficiency.	Temperatures, Degrees F.						
		Furnace Temperature.	Middle First Pass.	Top First Pass.	Top Second Pass.	Middle Second Pass.	Middle Third Pass.	Flue.
117.3	78.5	2336	866	655	619	511	471	455
126.7	79.6	....	893	689	646	526	485	473
128.6	79.8	2420	888	681	633	521	481	468
131.0	77.1	2455	889	682	642	519	479	468
131.0	75.3	2430	913	694	655	512	486	473
137.4	77.6	....	956	723	660	547	512	492
142.2	76.5	....	939	700	634	523	493	475
185.3	72.7	2530	1051	751	700	578	541	519

84. Cost of Boilers and Settings. — Figures giving the cost of boilers, irrespective of type, at so much per horse power are misleading, since the cost does not increase in the same ratio as the power. The wide variation in cost on the horse-power basis is partly due to the difference in rating. For instance, Scotch-marine boilers are ordinarily rated at 8 square feet of heating surface per horse power and return tubular boilers at 12 square feet. The price approximates one dollar per square foot of heating surface for all boilers over 100 horse power. The cost of water-tube and fire-tube boilers may be roughly estimated by the following formulas (C. H. Benjamin, Engr. U. S., Nov. 15, 1902):

(A) Cost in dollars = 500 + 9.2 × rated horse power.
(B) Cost in dollars = 500 + 8.5 × rated horse power.
(C) Cost in dollars = 100 + 6.5 × rated horse power.
(D) Cost in dollars = 100 + 5.0 × rated horse power.

(A) Horizontal water-tube boilers, 125 pounds pressure, 10 square feet heating surface per horse power.
(B) Vertical water-tube boilers, other conditions as in (A).
(C) Horizontal return tubular boilers, 12 square feet heating surface per horse power.
(D) Small vertical fire-tube boilers.

The cost of Scotch-marine boilers rated on a basis of 8 square feet per horse power may be estimated by means of formula (A).

The cost of plain settings may be roughly approximated as follows:

Horizontal water tube.

Cost = 400 + 0.8 × rated horse power.

Return tubular:

Cost = 300 + 0.7 × rated horse power.

85. Selection of Type. — Boilers constructed by builders of good repute are usually designed for safety, durability, and capacity, and rigid specifications and inspection of material and workmanship are ordinarily not necessary, as the makers' reputations are sufficient guarantee of their worth. Marked departure from standard designs must necessarily be specified, but in most cases instructions are limited to the extent of heating and grate surface, the character of the furnace, and arrangement of setting. Numerous tests on various types of boilers show practically the same efficiency provided the furnaces and boilers are properly designed, so that the relative merits may be considered with reference to (1) durability; (2) accessibility for repairs; (3) facility for cleaning and inspection; (4) space requirements; (5) adaptability to the type of furnace and stoker desired; (6) capacity; and (7) cost of boiler and setting. For high pressure, 150 pounds per square

inch or more, the water-tube or some form of internally fired boiler in which the shell plates are not exposed to the high temperature of the furnace is considered safer than the horizontal tubular boiler because the shell plates and the seams of the latter must be of considerable thickness in large units, and being exposed to the hottest part of the fire are likely to give trouble, especially if the water contains scale or sediment-forming elements. Return tubular and stationary locomotive boilers are seldom made in sizes over 200 horse power and hence are not to be considered for large units. For sizes over 150 horse power where overhead room is limited the return tubular boiler is most commonly installed, unless high pressure is essential, in which case the internally fired Scotch-marine boiler is peculiarly adaptable. The water-tube boiler is usually employed in large central stations for high-pressure units of 300 to 2500 horse power. The particular type of water-tube boiler is to some extent a matter of personal taste on the part of the engineer. For small powers and for intermittent operation, small vertical or horizontal fire-box boilers have the advantage of low first cost. The small air leakage and radiation losses give internally fired boilers an advantage over externally fired fire-tube or water-tube types, but this is partly offset by the greater extent of regenerative surface in the setting of the latter.* Internally fired boilers are more expensive than the externally fired, though the extra cost of setting and foundation in the latter may bring the total cost of the entire equipment to practically the same figure. The design and installation of the boilers and furnaces should be left at the outset to a capable engineer.

Makers usually request the following information from intending purchasers:

1. Steam pressure desired.
2. The quantity of steam demanded.
3. The kind of fuel to be burned.
4. The type of furnace or stoker.
5. The nature and intensity of draft.
6. Nature of setting.
7. Probable temperature of the feed water.

The complete specifications for a return tubular boiler are given in Chapter XIX.

86. Grates. — Grates may be divided into three general classes, namely, stationary, rocking, and traveling grates. The latter are treated in Chapter IV. Stationary grates are generally made of cast-iron sections in a variety of shapes as illustrated in Fig. 66. The bars

* At the power plant of the Cosmopolitan Electric Co., Chicago, the brick settings of the boilers (500 h.p. B. & W.) are completely incased with riveted boiler plates.

are ordinarily from 3 inches to 4 inches deep at the center (this makes them strong enough to carry the load caused by the weight of the fuel without sagging even when the top is red hot), $\frac{3}{4}$ inch wide at the top, and taper to $\frac{3}{8}$ inch at the bottom to enable the ashes to drop clear.

TABLE 33.

AIR SPACES AND THICKNESS OF GRATE BARS.

Size and Kind of Coal.	Width of Air Spaces. (Inch)	Thickness of Grate Bars. (Inch)
Screenings	$\frac{1}{4}$	$\frac{3}{8}$
Anthracite —		
Average	$\frac{1}{2}$	$\frac{3}{4}$
Buckwheat	$\frac{3}{8}$	$\frac{3}{8}$
Pea or nut	$\frac{1}{2}$	$\frac{1}{2}$
Stove	$\frac{5}{8}$	$\frac{1}{2}$
Egg	$\frac{3}{4}$	$\frac{1}{2}$
Broken	$\frac{7}{8}$	$\frac{1}{2}$
Lump	1	$\frac{3}{4}$
Bituminous, average	$\frac{5}{8}$	$\frac{3}{4}$
Wood —		
Slabs	$\frac{3}{4}$	$\frac{3}{4}$
Sawdust	$\frac{1}{4}$ to $\frac{3}{8}$	$\frac{3}{8}$
Shavings	$\frac{1}{2}$ to $\frac{5}{8}$	$\frac{1}{2}$

The width of the air space is determined by the size of the fuel to be used, the average proportions being given in Table 33.

The "Tupper" and "Herringbone" grate bars are stiffer and less likely to warp than the common form, but are not so readily sliced and

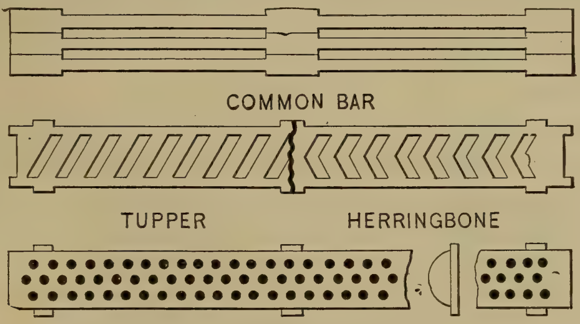


FIG. 66. Types of Grate Bars.

therefore not so convenient with coal that clinkers badly. Sawdust or pinhole grates are used in burning sawdust, tanbark, and very small sizes of coal. Grates are often set horizontally and the bars are held in place simply by their own weight, but long grates are best placed

sloping toward the rear to facilitate firing. The front of the grate when designed for bituminous coal is often made solid, this portion being called the "dead plate." It serves to hold the green fuel until the hydrocarbons have been distilled off, when the charge is pushed back on the open grate at the time of next firing. The length of a single bar or casting should not exceed three feet. The length of grate may be made of two or three bars and should not exceed 6 feet with bituminous coal, as this is the greatest length of fire that can be readily worked by a stoker. With buckwheat anthracite furnaces 12 feet in depth are not unusual, as anthracite fires require no slicing.

The disadvantage of using stationary grates is that the fire is not easily cleaned. Unless the air spaces are kept free of clinkers and ashes, combustion is hindered and the fire rendered sluggish. Frequent cleaning, however, is wasteful of fuel and reduces the furnace efficiency by letting in a large excess of air every time the fire door is opened.

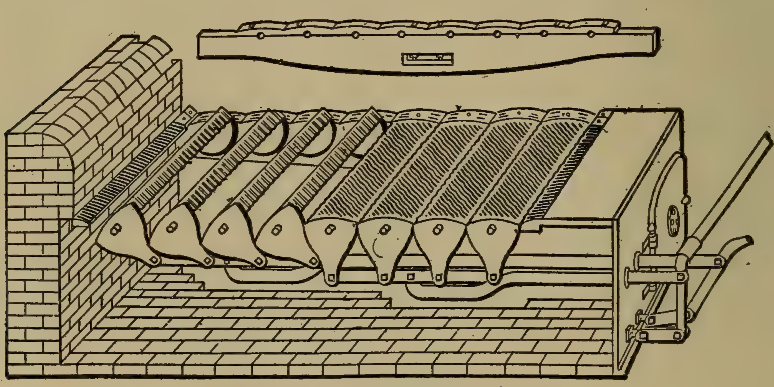


FIG. 67. A Typical Shaking Grate.

87. Shaking Grates. — Shaking grates have the advantage of permitting stoking without opening the fire door and require less manual labor than stationary grates. There is a great variety of sectional shaking grates on the market and some of them are made self-dumping. One of the best-known types is illustrated in Fig. 67. Each row or section of grate bars is divided into a front and a rear series by twin stub levers and connecting rods. An operating handle is adapted to manipulate either one or both of the levers in such a manner that the front and rear series may operate separately or together. The shaking movement causes no increase in the size of the openings and hence prevents the waste of fine fuel. Ordinarily the width of the grate is made equal to two or more rows of grate bars so that the live fire may be shoved sidewise from one row to the other when cleaning. A depth of

fire of from 6 to 10 inches is carried according to the nature of the fuel and the available draft.

Grate Bars: Engr. U.S., Nov. 1, 1906, p. 728, Jan. 1, 1907, p. 68; Am. Elecn., Jan. 1904, p. 269.

88. Blow-Offs. — Boilers must be provided with blow-off pipes for draining off the water and for discharging sediment and scale-forming material. The "bottom blow" is ordinarily an extra-heavy pipe of

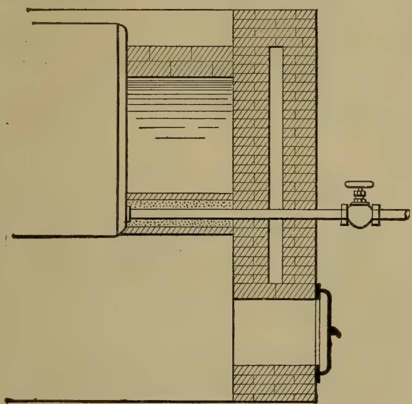


Fig. 68. Horizontal Blow-off Connection to Head.

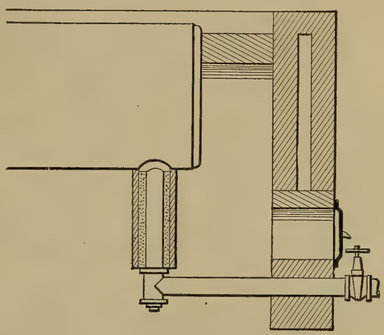


Fig. 69. Vertical Blow-off Connection to Shell.

suitable diameter connected to the lowest part of the boiler and fitted with a valve or cock, or both. (See Fig. 512.) Fig. 68 shows an arrangement of horizontal blow-off connected to the head of a return

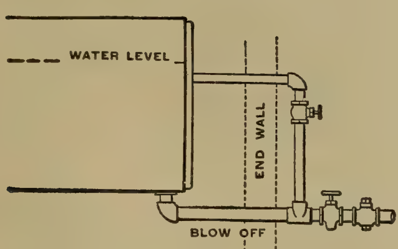


Fig. 70. Blow-off Connection with Circulating Pipes.

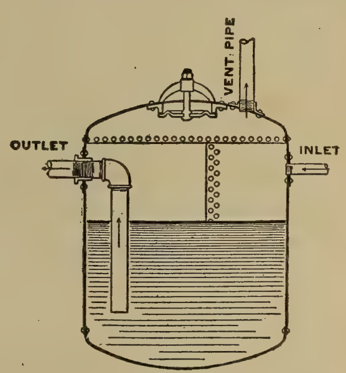


Fig. 71. Blow-off Tank and Connections.

tubular boiler and Fig. 69 a vertical blow-off connected to the shell. The latter is the better arrangement. The blow-off pipe where it passes through the back connection is covered with magnesia, asbestos,

or fire brick. When exposed to the action of extremely hot furnace gases as in forced-draft installations, the arrangement illustrated in Fig. 70 is sometimes used to prevent the pipe from burning out. When

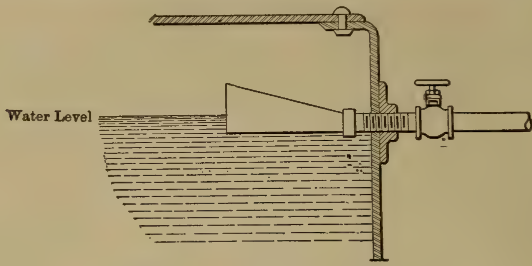


FIG. 72. Surface Blow-Off.

the blow-off cock is shut and the valve on the vertical branch is open there is a continuous circulation of water. Where boilers are arranged in batteries, the battery may have a common outlet for the blow-off pipes, as illustrated in Fig. 512. Usually the blow-off pipes may discharge into the open air, but this is not permissible in large cities, nor is it lawful to blow directly into the sewer. In this case the water and sediment may be discharged into a *blow-off tank* and permitted to cool before delivery to the sewer, as illustrated in Fig. 71.

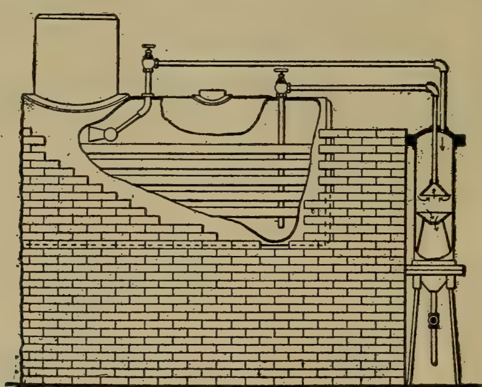


FIG. 73. Buckeye Skimmer.

"*Surface blows*" are often installed to remove scum, grease, and floating or suspended particles of dirt. The bell-mouthed shape shown in Fig. 72 permits the skimmer to accommodate itself to varying water level, and it is sometimes provided with a float and with a flexible joint, Fig. 89.

89. Damper Regulators.— For maximum furnace efficiency the draft must be regulated to burn just enough fuel to supply the steam

required. Where forced draft is employed this is done by regulating the speed of the blower. With natural draft it is the usual practice to regulate the draft by means of dampers placed in the uptake, and in order that the regulation may be effective it should be automatic. Automatic dampers are economical and useful and are particularly desirable in plants where the demand for steam fluctuates rapidly. There are several successful types on the market, some operated by

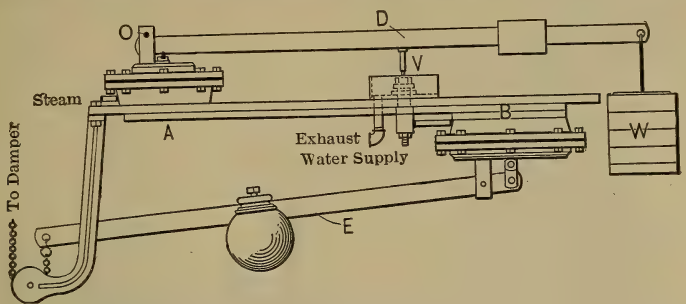


FIG. 74. Kitts Hydraulic Damper Regulator.

water pressure and others by direct boiler pressure. Fig. 74 illustrates such a mechanism. Full boiler pressure acting at all times on a diaphragm *A* raises or lowers a weight *W* attached to arm *D* according as the steam pressure increases or decreases. Arm *D* actuates a small valve *V* which controls the supply of water to chamber *B*. A diaphragm in chamber *B* raises and lowers the damper as the water pressure varies, a drop of 0.5 pound being sufficient to open the damper to its maximum. The steam diaphragm has a movement of only 0.01 inch and the water diaphragm 0.5 inch. When properly adjusted and given proper attention automatic dampers work in a very satisfactory manner.

Fig. 75 shows a section through the Tilden damper regulator, illustrating the principles of the steam-actuated type. The device is connected directly to the boiler by pipe *A*. The pressure on piston *B* is balanced by spring *C* under normal conditions of operation. Any variation from the normal will cause the rod *R* to move up or down, so that the dampers are opened or closed in proportion to the change in pressure. The chamber *N* is separate from chamber *M*, so that steam or water cannot come in contact with the spring. Piston *D* acts as a guide only. In a recent design of this device the regulator is hydraulically actuated and simultaneously operates the damper and the stoker engine thereby automatically proportioning the air and fuel supply to the load requirements.

90. Water Gauge. — The water level in a boiler is usually indicated either by a gauge glass, by try cocks, or both, connected directly to the

boiler as in Fig. 1, or to a *water column* or *combination* as in Fig. 76. Each gauge-glass connection should be fitted with a stop valve which may be closed in case the tube breaks. In large boilers these valves, usually of the quick-closing type, are conveniently operated from the boiler-room level by means of a chain attached to the valve stem. Self-closing automatic valves are frequently employed, one type being illustrated in Fig. 77. If the glass breaks the outrush of steam forces

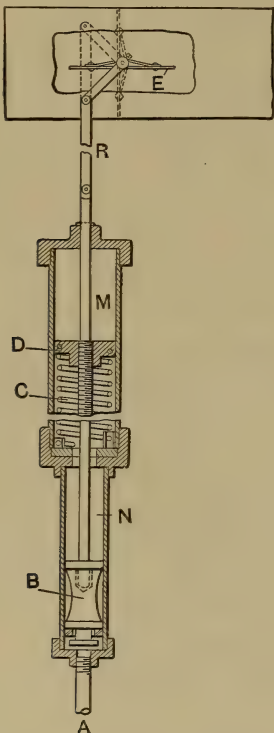


FIG. 75. Tilden Steam-actuated Damper Regulator.

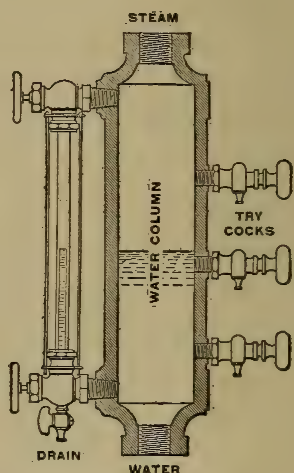


FIG. 76. Simple Water Column.

the ball against a conical seat and shuts off the supply. When a new glass is inserted the ball is forced back by slowly screwing in the valve stem. Hinged valves mechanically operated from without are considered more reliable than ball valves, as scale is less likely to render them inoperative.

Try cocks or *gauge cocks* are set at points above and below the desired water level, preferably connected directly to the boiler shell, but sometimes to a water column as in Fig. 76. The water level is ascertained by opening the cocks in succession.

The objection to the latter arrangement is that accident to or a stoppage of the piping renders both gauge glass and try cocks useless. Water columns should be blown out once a day, and the gauge cocks opened to see that the height of the water indicated tallies with that shown by the glass. Some engineers prefer two separate columns to each boiler and no cocks, others rely solely upon cocks.

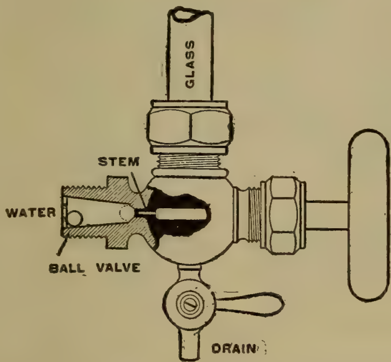


FIG. 77. Water Gauge with Self-closing Valve.

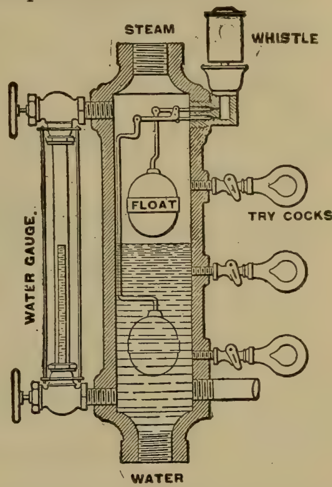


FIG. 78. Combined Water Column and High- and Low-water Alarm.

The water column shown in Fig. 78 has an alarm whistle, controlled by two floats, which gives a high- and low-water alarm. Numerous devices of this class are on the market but they are usually regarded as unreliable and most engineers are content to depend upon water gauge and try cocks.

Water Gauges and Columns: Mach., Sept., 1905, p. 31; Power, Aug., 1905, p. 483; Am. Elecn., July, 1904, p. 359; Engr. U. S., Jan. 1, 1907, p. 58.

91. Fusible or Safety Plugs. — Fusible or safety plugs as illustrated in Fig. 79 are brass plugs provided with a fusible metal core. They

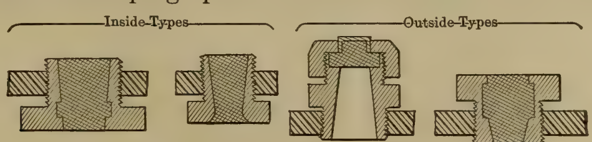


FIG. 79. Types of Fusible Plugs.

are inserted in the shell or tubes at the lowest permissible water line. When covered by water the heat is conducted away sufficiently fast to keep the temperature below the fusing point, but when uncovered the low conductivity of the steam prevents the rapid withdrawal of heat,

whereupon the alloy melts and the blast of escaping steam gives warning. The melting point of fusible metals being sometimes uncertain, plugs occasionally blow out without apparent cause and at other times fail to act when shell is overheated. Fusible plugs are required by law in many cities.

92. Mechanical Tube Cleaners. — Although purifying plants, boiler compounds, and the like are preventive of scale formation to a great extent, experience shows that the most satisfactory method is to use mechanical tube cleaners for cutting or breaking the scale. The principles of construction of these devices vary widely according to the types of boilers in which they are used, and depend upon the nature of

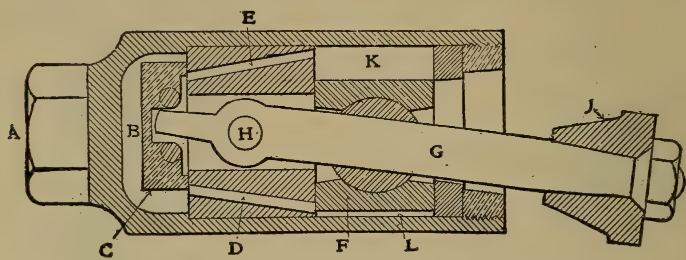


Fig. 80. Mechanical Tube Cleaner — Hammer Type.

the duty which they must perform. They may be conveniently divided into two classes:

1. Those which loosen the scale by a series of rapid hammer blows, Fig. 80.
2. Those which cut out the scale by a revolving tool, Fig. 81.

The hammer device is applicable to either the water- or fire-tube type of boiler, but the revolving cutter is applicable to the water-tube only. Steam, compressed air, or water under pressure may be used as the motive power, though the latter is the most convenient and satisfactory.

Referring to Fig. 80, the hammer head J is given a rapid motion, which may reach 1500 vibrations per minute, and subjects the tube to repeated shocks, thereby cracking the brittle scale and jarring it loose from the water surface of the tube. The cleaner head is attached to a flexible pipe of sufficient length to enable it to be pushed from one end to the other. Even if carefully manipulated the hammer is apt to injure the tube by swaging it to a larger diameter, producing crystallization in the metal and causing leaks where the tubes are expanded into the sheets, hence its use is not to be recommended.

Hydraulic turbine cutters are made in many designs, one of which is shown in Fig. 81. The cylindrical casing D contains a hydraulic turbine consisting of a fixed guide plate which directs the water at the

proper angles upon the vanes of the turbine wheel T. The cutters C revolve at high speed and chip the scale into small pieces. The stream of water flowing from the turbine envelops the cutters, keeps their edges cool, and washes away the scale as fast as it is detached. Different styles of cutter wheels are furnished with each cleaner so as to adapt the device to all kinds of scale formations. In well-managed plants scale is not permitted to deposit to a thickness greater than $\frac{1}{16}$ to $\frac{3}{16}$ of an inch.

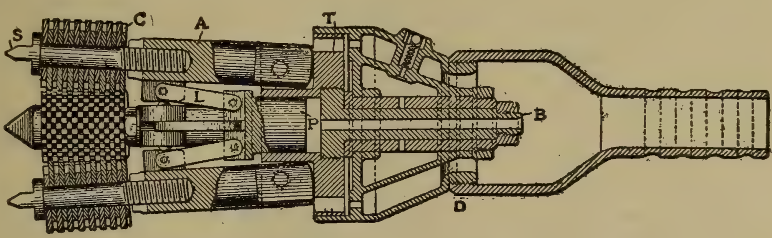


FIG. 81. Mechanical Tube Cleaner — Turbine Type

The soot and cinders which accumulate on the inside surface of fire-tube boilers are removed by mechanical scrapers, brushes, or steam-jet blowers. (For a description of these devices see Power, Dec. 6, 1910, p. 452.)

The tubes of a water-tube boiler are cleaned externally by means of a steam jet.

Arches, Firebrick Furnaces: Power, Feb. 20, 1912.

Blow-off Connections: Locomotive, Oct., 1906; Elec. World, Nov. 2, 1907; Nat. Engr., June, 1904; Eng. Rec., May 9, 1908; Steam, Feb., 1911.

Bracing: Boiler Maker, Feb., 1912; Mach., Sept., 1903, p. 18, Oct., 1903, p. 83; Power, Jan., 1903, p. 24, Oct., 1905, p. 611, Nov., p. 687; Eng. News, Dec. 15, 1904, p. 533; Engr. U. S., Jan. 1, 1907, p. 18; Prac. Engr., Jan., 1907.

Boiler Cleaning: Am. Elecn., Dec., 1900, April, 1904, p. 174; Power, May and Oct., 1905, Aug., 1906, p. 465; Locomotive, Oct., 1904; Boiler Maker, Aug., 1905; Engr. U. S., Jan. 1, 1907, p. 109; Power & Engr., Nov. 8, 1910, p. 1975.

Boiler Design: Engr. U. S., Jan. 15, 1902, p. 59; Eng. Mag., May, 1904, p. 233; Eng. Rec., July 14, 1900, May 18, 1901, p. 467, Oct. 12, 1901, p. 347; Power, Oct., 1901, p. 14, March, 1906, p. 147; Am. Mach., April 21, 1904.

Boiler Dimensions: All Types of Stationary Boilers: Eng. U. S., Jan. 1, 1907, p. 10, Aug. 1, 15, 22, 1903. *Small Marine Boilers:* Am. Mach., Sept. 3, 1896, p. 823. *Tubular Boilers:* Mach., Oct., 1902, p. 94.

Circulation in Boilers: Eng. Rec., July 20, 1901; Cassier's Mag., Jan., 1905; Elec. Rev., Lond., April 4, 1902; Engng., April 18, 1902; Engr., Lond., Nov. 6, 1903; Am. Mach., Jan. 14, 1897, p. 40, Sept. 20, 1900, p. 910; Eng. News, Jan. 18, 1900, p. 40; Trans. A.S.M.E., 7–814, 9–489; Engr. U. S., Oct. 15, 1907.

Damper Regulators: Engr. U. S., Jan. 1, 1907, p. 58; Elec. Wld., May 2, 1908.

Domes: Engr. U. S., Jan. 1, 1907, p. 27.

Classification of Boilers and Comparison of Types: Engr. U. S., Jan. 1, 1907; Min. Rept., Feb. 21, 1907.

Explosions, Cause of: Power, Mar. 26, 1912; Boiler Maker, July, 1910.

Furnace and Settings: Power & Engr., Jan. 3, 1911, p. 2; Elec. World, Sept. 7, 1907; Am. Elecn., Jan., 1902, p. 10, Nov., 1903, p. 557, July, 1904, p. 339; Engr. U. S., July 15, 1905, p. 471, Sept. 15, 1905, p. 622, Aug. 1, 1906, p. 491, May 15, 1906, Jan. 1, 1907; Power, March 24, 1908, p. 445, June, 1905.

Inspection: Power, Jan., 1906, p. 32; Engr. U. S., Feb. 15, 1907; Trans. A.S.M.E., 4–142; Boiler Maker, Feb., 1911.

Riveted Joints: Power, March, 1906, p. 147, April, 1906, p. 227; Engr. U. S., Jan. 1, 1907, p. 21, Aug. 15, 1907, p. 784; Boiler Maker, June, 1906, Dec., 1907; Prac. Engr., Dec. 13, 1907.

Safety Valves: See paragraph 390.

Specifications: Power, Dec., 1905, p. 728; Nat. Engr., May 15, 1903, p. 367; Boiler Maker, Sept., 1906, p. 243.

Thickness of Boiler Plate: Am. Mach., Jan. 16 and Feb. 27, 1902; Trans. A.S.M.E., 22–127, 15–629, 24–921; Eng. News, Jan. 31, 1901, p. 121.

Testing: See A.S.M.E. Code for conducting Standard Boiler Trials, reprinted in Appendix B. See also Power and Engr., Feb. 23, 1909.

Testing to Destruction: Eng. News, Feb. 1, 1912.

Bridge Walls in Theory and Practice: Power and Engr., Mar. 9, 1909, p. 452.

Soot Blowers and Soot Suckers: Power and Engr., Dec. 6, 1910, p. 2143.

Installing Tubes in Boilers: Power and Engr., Nov. 1, 1910.

CHAPTER IV.

SMOKE PREVENTION, FURNACES, STOKERS.

93. Current practice shows that bituminous coals high in volatile matter can be efficiently burned without smoke, provided the furnaces are properly designed, and the necessary attention is given draft requirements and stoking.

The problem of smoke abatement is a comparatively simple one for large plants equipped with mechanical stokers and provided with ample draft, even for widely fluctuating loads, but for hand-fired plants it depends largely upon skillful manipulation by interested and efficient firemen. The order of intelligence demanded for this work is out of all proportion to the wages paid. In many small plants — and these are usually the most obstinate smoke offenders — the fireman handles as much as a ton of coal per hour by hand, besides caring for the feed pumps and water levels, keeping the boilers clean, and removing the ash. The boiler room is frequently poorly lighted and poorly ventilated. It is, therefore, not surprising that the fireman seldom worries about the smoke problem. A better wage scale and more consideration for the fireman might do a great deal toward abating the smoke nuisance.

Since the loss in heat due to *visible* smoke is usually less than one per cent, and seldom greater than two per cent of the heat value of the coal, it is a common statement among owners of power plants that it is cheaper to smoke than to operate without smoke. This is undoubtedly true in many cases where smokeless combustion can be secured only by admitting a considerable excess of air with a consequent loss in economy frequently greater than that due to incomplete combustion and smoke, but if proper attention is given to the various factors involved practice shows that smokeless combustion can be effected with high boiler and furnace efficiency.

In order that combustion may be smokeless and efficient, the volatile gases and separated free carbon must be brought into intimate contact with the proper quantity of air and maintained at a temperature above the ignition point until oxidation is complete before they are brought into contact with the heat-absorbing surfaces of the boiler. Mere excess of air will not effect smokeless combustion, even if the gases and air are thoroughly mixed, if the temperature is prematurely reduced below that neces-

sary for combustion by contact with the heat-absorbing surfaces of the boiler.

Smoke may be produced, therefore, by

1. An insufficient amount of air for the perfect combustion of the volatile gases. This is primarily a function of the draft.

2. An imperfect mixture of air and combustible.

3. A temperature too low to permit complete oxidation of the volatile combustible.

The curves in Fig. 82 give an idea of the distribution of smoke production by various industries in Chicago, in 1910. Rigid enforcement of the smoke ordinance has reduced the nuisance to a considerable extent, so that the distribution at this date differs somewhat from that shown in the curves.

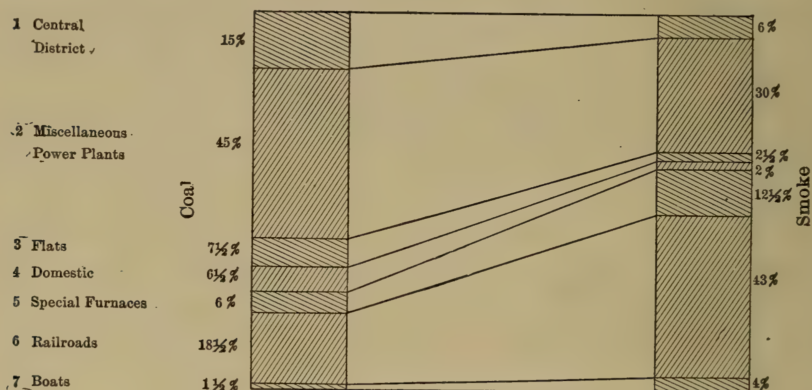


FIG. 82. Smoke Distribution in Chicago Plants.

Smoke-preventing devices may be divided into two classes: (1) those which may be conveniently attached to plants already in operation without material modification of the furnace, such as steam jets and other means of mixing the air and combustible gases, admission of air through the bridge or side wall, and mechanical draft; and (2) those which are an integral part of the boiler and setting, such as mechanical stokers, Dutch ovens, down-draft furnaces, and fire-tile combustion chambers incorporated with the regular setting.

94. Hand-fired Furnaces. — Hand-fired furnaces, as a class, are most obstinate smoke producers. Although they can be operated efficiently without the production of objectionable smoke the result depends more upon the fireman than upon the design of the furnace. The chief difficulty with hand-fired furnaces lies in the intermittent nature of the firing. When a fresh charge of coal is fed into the furnace an enormous volume of volatile matter is evolved. For complete combustion a

corresponding amount of air must be supplied and intimately mixed with the volatile gases before contact is made with the comparatively cool heating surface. In the average hand-fired furnace the combustion chamber is so small and the heating surface is so close to the grate that the partly burned gases strike the heating surface before oxidation is complete and combustion is hindered or even completely arrested. The majority of so-called smoke preventers are merely devices for mechanically mixing the air and volatile gases. These include fire-brick piers, baffles, arches, and steam jets. There is no question as to the value of these mixing devices if properly installed, but the personal element is too variable a factor for consistent results and the ultimate solution lies in mechanical stoking. The most economical and smokeless hand-fired plants are those that approach the continuous feed of the mechanical stoker. The following rules formulated by the Coal Stoking and Anti-Smoke Committee of the Illinois Coal Operator's Association for firemen using Illinois and Indiana coal in hand-fired furnaces give some idea of the principles involved in effecting smokeless combustion:

1. Break all lumps and do not throw any in furnace any larger than one's fist. The reason for this is, that large lumps do not ignite promptly and their presence also causes holes to form in the fire, which allow the passage of too much air.

2. Keep the ash pits bright at all times. If they become dark it is evident that the fire is getting dirty and needs cleaning, which, if not done, will cause imperfect combustion and smoke. If the furnace is equipped with a shaking grate, it should be operated often enough to prevent any accumulation of ashes in the fire. Do not allow ashes to collect in the ash pits, as they not only shut off the air supply, but may cause the grate to be burned.

3. In firing do not land the coal all in one heap, but spread it over as wide a space as possible as it leaves the shovel. A little practice will enable one to catch the proper motion to give the shovel to make the coal spread properly.

4. Place the fresh coal from the bridge wall forward to the dead plate and do not add more than 3 or 4 shovels at a charge. If this amount makes smoke it should be reduced till smoke ceases, which means, of course, that firing will be at more frequent intervals than formerly to keep up steam. This rule applies in cases where the boiler is worked at a large capacity. In such instances, however, where a small capacity only is required, firing by the coking method is the best, wherein the fresh coal is placed at the front of the fire and pushed back and leveled when it has become coked.

5. Fire one side of the furnace at a time so that the other side containing a bright fire will ignite the volatile gases from the fresh charge.

6. Do not allow the fire to burn down dull before charging. If this is done, it will not only result in a smoky chimney, but an irregular steam pressure.

7. Do not allow holes to form in the fire. Should one form, fill it by leveling and not by a scoop full of coal. Keep the fire even and level at all times. As far as possible level the fire after the coal has become coked.

8. Carry as thick a fire as the draft will allow, but in deciding on the proper thickness, judgment must be exercised. If the draft is poor, a thin fire will be in order, but if strong, a thicker fire should be carried.

9. Regulate the draft by the bottom or ash-pit doors and not by the stack dampers, because when the stack damper is used it tends to produce a smoky chimney, as it reduces the draft, while the closing of the ash-pit door diminishes the capacity to burn coal. If strict attention is given to firing, and accounting to demand for steam, there will be no occasion to have recourse to dampers, except when there is a sudden interruption in the amount of steam being used.

10. A good general rule is to fire little and often, according to steam demands, rather than heavy and seldom. The former means economy in fuel and a clean chimney, while the latter signifies extravagance in fuel and a smoky chimney.

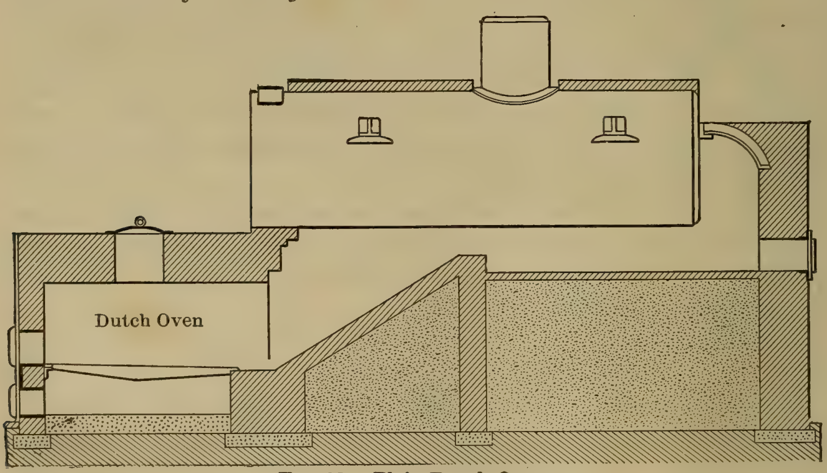


Fig. 83. Plain Dutch Oven.

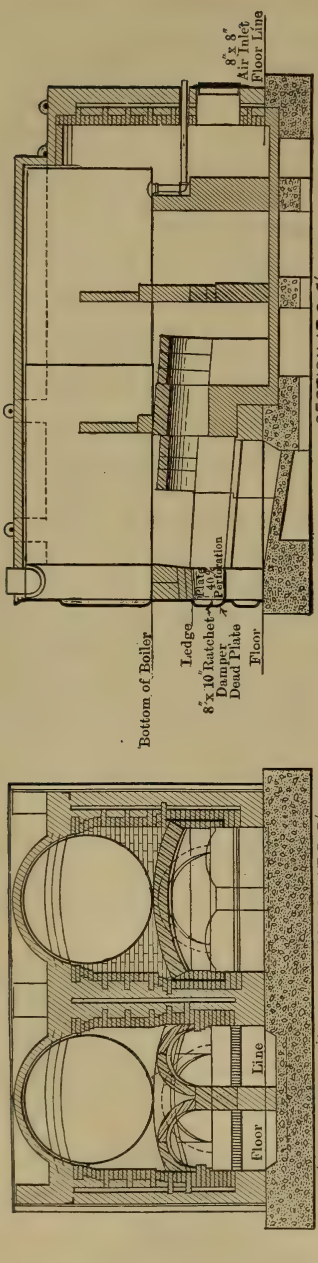
95. Dutch Ovens. — An independent furnace or Dutch oven in front of the boiler as illustrated in Fig. 83 provides one of the simplest methods of securing a large combustion chamber for the mingling of

the air and combustible gases before delivering them to the boiler proper. Such a furnace produces very high temperatures when operating under best conditions, and hence must be lined with fire brick of excellent quality. Although better than the ordinary setting the plain Dutch oven is too limited in length and capacity to prevent smoke from forming, except at very light loads. The velocity of the gases is usually too high to permit either a thorough mixture or complete oxidation before striking the boiler tubes. Steam jets placed at the sides of the setting and blowing across the fire are sometimes effective in mixing the air and combustible gases, but the best results are obtained by modifying the construction of the furnace to the extent of introducing deflecting arches which vary the direction of flow and by increasing the length of the path of the heated gases. The greater the length of the path and the greater the number of baffles the more thoroughly will the air and gases be mingled, but the intensity of draft will of course be decreased in proportion. A compromise must therefore be made between required draft and length of path. The larger the extent of fire-tile surface the greater will be the regenerative effect, which is of particular importance in hand firing when the evolution of volatile gas is intermittent, but the first investment and cost of repairs and renewals are greater. There are little reliable data available pertaining to the relation between capacity of furnace and length of path of the heated gases for maximum efficiency. A modified Dutch oven is illustrated in Fig. 84. The extension front is not necessary with all types of boilers, as will be seen from Figs. 86 and 87, in which a tile roof and baffles suitably arranged within the setting proper simulate the Dutch-oven effect.

96. Twin-fire Furnace. — This arrangement, illustrated in Fig. 84 in connection with a hand-fired return tubular boiler, is a double furnace formed by longitudinal arches extending between bridge wall and fire door.

The furnaces are fed and manipulated alternately, the object being to have one furnace in a highly incandescent state, while green fuel is fed into the other. Air is admitted both below and above the grate, and the volatile gases are supplied with the necessary oxygen for combustion before they come into contact with the comparatively cool boiler surface.

The gases from both furnaces first pass into a chamber formed by a single arch sprung across the entire inner setting from the side wall, a short retarding arch being placed between this intermediate chamber and the rear of the setting. A special tile of high-grade refractory clay is used, the thickness varying from 4 to 6 inches, depending upon the



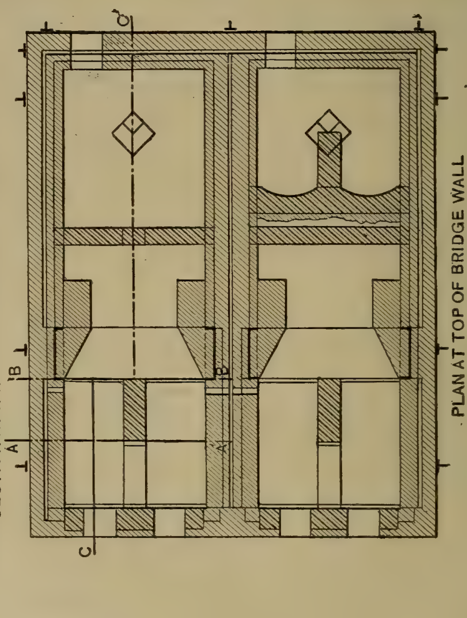


FIG. 84. "Twin Fire Arch" Applied to Two Return Tubular Boilers.
(Chicago Specifications.)

size of furnace and the length of span. The furnace can readily be substituted for the ordinary types in common use under any standard tubular or water-tube boiler and may be installed either under the boiler, as indicated in the illustration, or in an extension Dutch oven. This is an excellent furnace, and when properly manipulated gives smokeless and efficient combustion.

97. Chicago Settings for Hand-fired Return Tubular Boilers. — Figs. 86 and 87 show the general details of settings for return tubular boilers as recommended by the Chicago Department of Smoke Inspection. The setting illustrated in Fig. 85, and known as No. 7, is ordinarily installed on heating jobs where the rate of combustion is 12 pounds of coal per square foot of grate surface per hour or less, and those shown

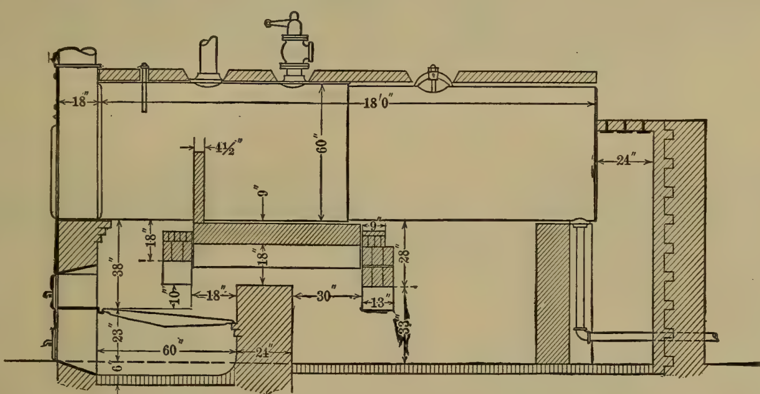


FIG. 85. Smokeless Setting for Hand-fired Return Tubular Boiler. (Chicago Specifications.)

in Figs. 86 and 87 where the rate of combustion is greater or where the plant has a regular power load. The dimensions in Fig. 85 refer to a specific set of conditions and are not general. All three settings require careful manipulation for smokeless combustion as is the case with hand-fired furnaces in general. It has been the experience of the Department that most violations of the smoke ordinance are due primarily to insufficient draft, the required rate of combustion being too high for the available air supply. The requirements outlined in paragraph 93 apply equally well to these settings. The following specifications refer to Figs. 85, 86, and 87, the items in the specifications corresponding to the letters in the illustrations.

A. Doors should be of a type allowing the admission of excess air over the fire when so desired. If panels are cut in the fire doors for this purpose, the aggregate area of the openings should be not less than 4 square inches to each square foot of grate surface.

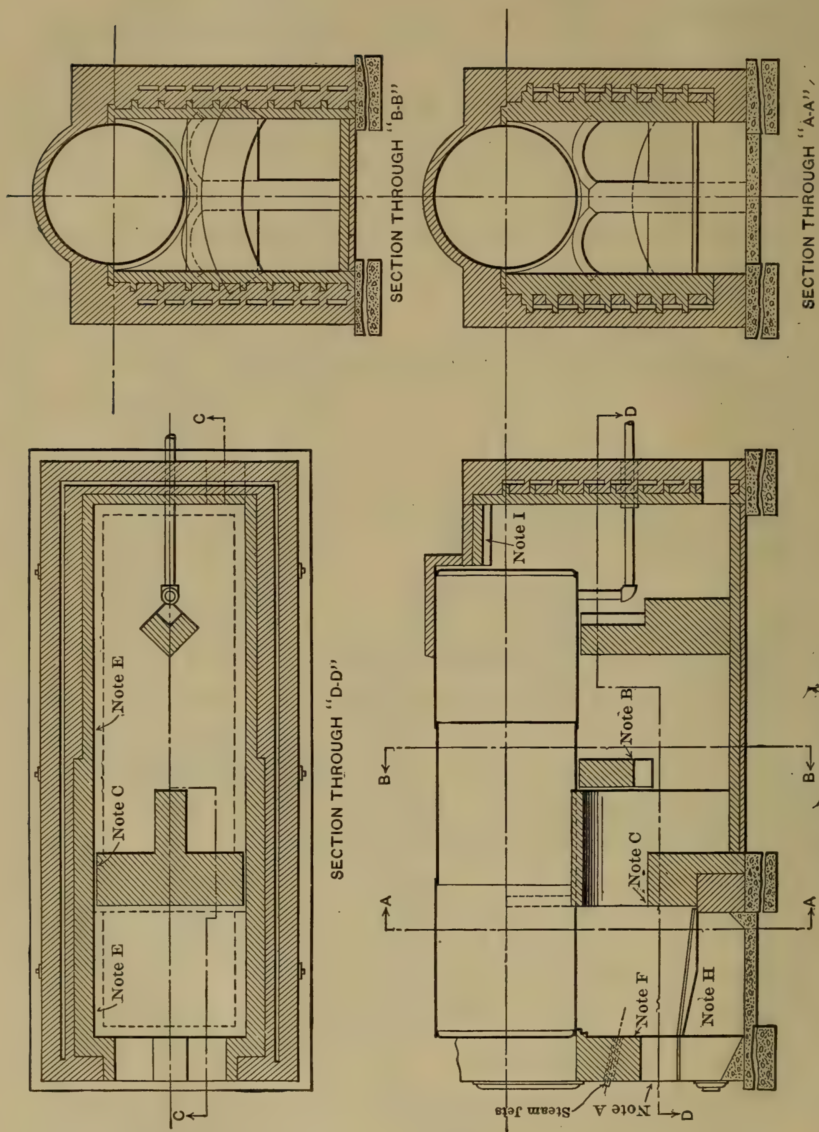


FIG. 86. Smokeless Setting for Hand-Fired Return Tubular Boiler with Double Arch Bridge-Wall. (Chicago Specifications.)

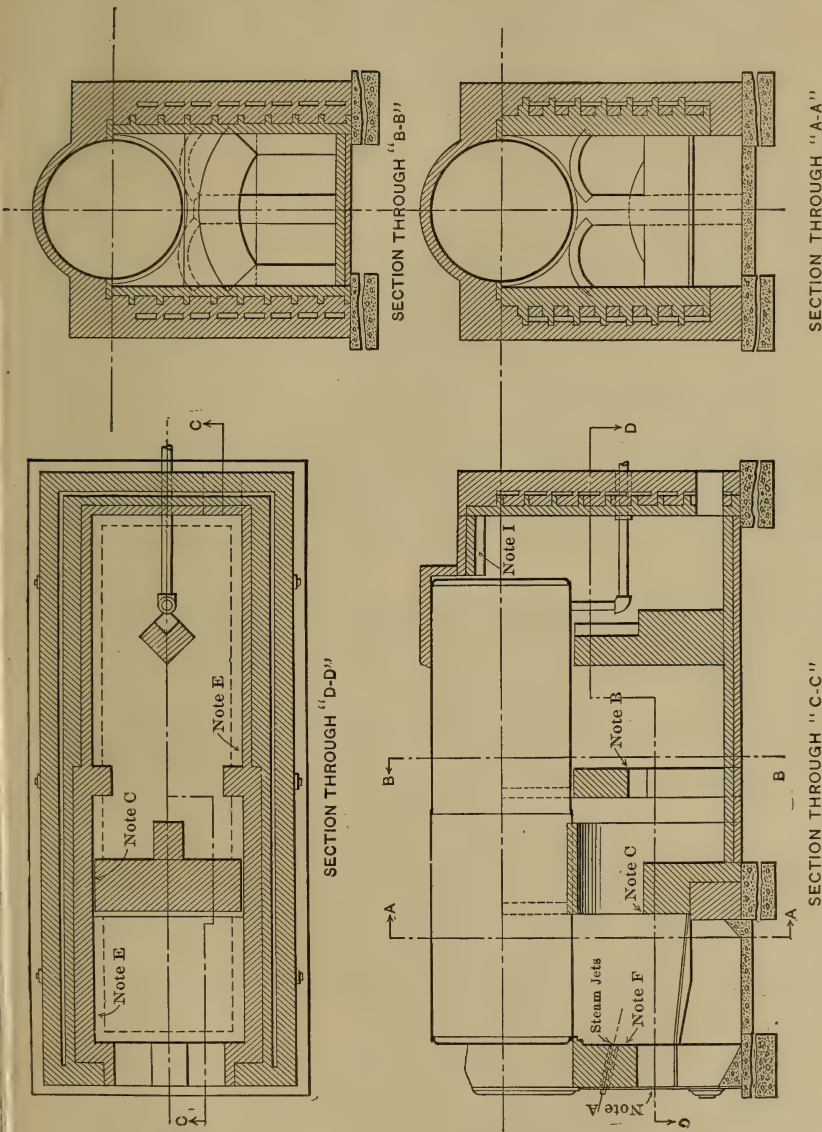


FIG. 87. Smokeless Setting for Hand-Fired Return Tubular Boiler with Double Arch Bridge-Wall. (Chicago Specifications.)

B. Arches should be made of wedge brick or "bull heads" and not laid in two courses of $4\frac{1}{2}$-inch brick.

C. The bridge wall should be made of first-grade fire brick above the grate line and with fire-brick facing not less than 9 inches in thickness on the combustion-chamber side. The top row should be a row-lock course. Provision should be made in the building of the bridge wall for lateral expansion.

D. The combustion chamber floor should be paved with fire brick laid on edge.

E. Fire-brick lining below the arch skewbacks should be not less than 9 inches in thickness. Fire-brick lining above the arch system and behind the deflection arch may be $4\frac{1}{2}$ inches in first-grade fire brick, with headers every fifth row.

F. Fire brick over firing-door liners should be arched. This rule also applies to brick above the clean-out door openings.

G. Facilities for taking up arch thrust should be provided in every case by suitable metal reënforcements extending horizontally throughout the length of the arches. No air space should intervene between the metal reënforcements and the skew-backs.

H. Herringbone or Tupper grates or other similar types should not be selected where bituminous coal forms the major portion of the fuel.

I. The back arch is preferably sprung from side to side rather than from back wall to rear boiler tube sheet. No metal should be exposed to direct heat of gases.

J. Chimney heights of less than 90 feet above the grate line should not be permitted, and this height allowed only when the chimney is direct connected to the boiler uptake. In case of a breeching and detached chimney, add to the height of chimney computed by standard methods (never less than 75 feet) 10 feet for every turn of the breeching and one foot for each foot in length of the breeching.

K. For boilers of all sizes, special provision for the examination of girth seams must be made.

L. In the event of arch failures, the boiler should be immediately taken out of service. This is to avoid failure of the boiler shell due to heat being applied upon a portion of the heating surface over which a mud deposit has formed.

98. Wooley Smokeless Furnace. — Fig. 88 shows a longitudinal section and a sectional plan of a Wooley smokeless furnace applied to a B. & W. boiler. The main features of the furnace are a dividing wall in the fire box and a deflecting wall in the combustion chamber. The dividing wall permits of the alternate method of firing, whereby one side of the furnace is always in an incandescent state while the other

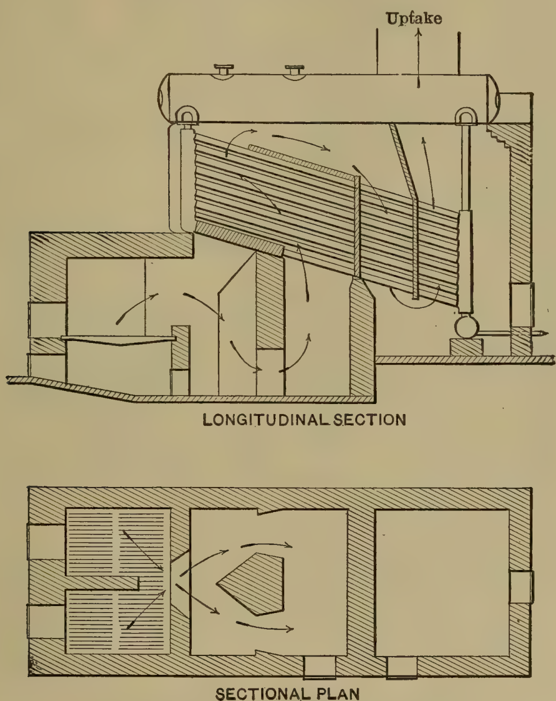


Fig. 88. Wooley Smokeless Furnace.

side is being supplied with green fuel. If a mechanical stoker is used the wall in the fire box is omitted. The products of combustion are intended to be thoroughly mingled with the requisite amount of air by the deflecting walls before entering the regenerative or secondary combustion chamber. This type of furnace does not meet Chicago requirements and is not much in evidence in the middle West.

99. Kent's Wing-wall Furnace. — Fig. 89 shows the application of Kent's wing-wall furnace to a water-tube boiler. The Dutch oven in front of the regular setting contains the grates. Wing walls *E* are placed as shown two or three feet to the rear of the bridge wall *D*, and fire-brick piers *H* behind the wing walls.

In operation, fresh coal is spread alternately over each half of the grate. The dense smoky gases which rise from the green portion of the fire mingle in the narrow passage with the highly heated air which comes through the other side of the grate greatly in excess of that

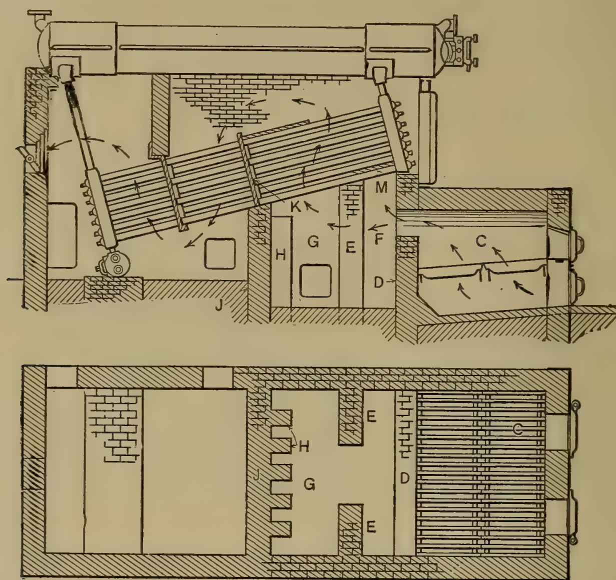


Fig. 89. Kent's Wing-wall Furnace.

required to consume the partially burned coal there. The piers *H* act as regenerative surfaces, absorbing heat from the fire when it is hottest and giving it out when it is coolest, that is, just after firing. Wing-wall furnaces are little used in the middle West.

100. Burke's Smokeless Furnace. — Figs. 90 and 91 show sections through a Burke smokeless furnace as installed in a number of tall office buildings in Chicago. It amounts virtually to a Dutch oven equipped with shaking grates, and embodies an extension self-feeding coking oven of cast-iron section lined with fire brick and protected from overheating by air circulation through the sections. Natural draft is used, the fire doors being closed; but air is admitted above as well as below the fire. As this stoker is manipulated by hand, more or less attention is required of the operator in keeping the fire clean. Furnaces of this type at the power plant of the Majestic Theatre building, Chicago, Ill., are giving excellent results.

101. Down-draft Furnaces. — Fig. 92 shows the application of a Hawley down-draft furnace to a Heine water-tube boiler. In this furnace there are two separate grates, one above the other, the upper

one being formed of parallel water tubes connected with the water space of the boiler through the steel headers or drums, *A* and *D*, in such a manner as to insure a positive circulation. Fuel is supplied

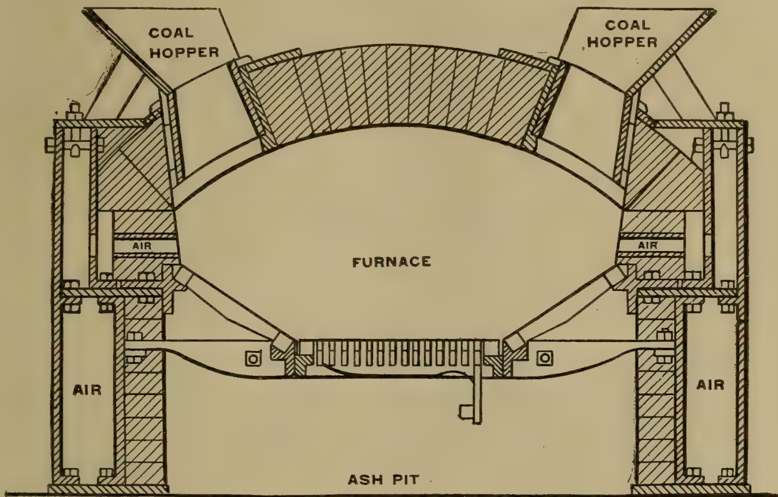


FIG. 90. Burke's Smokeless Furnace — Front Section.

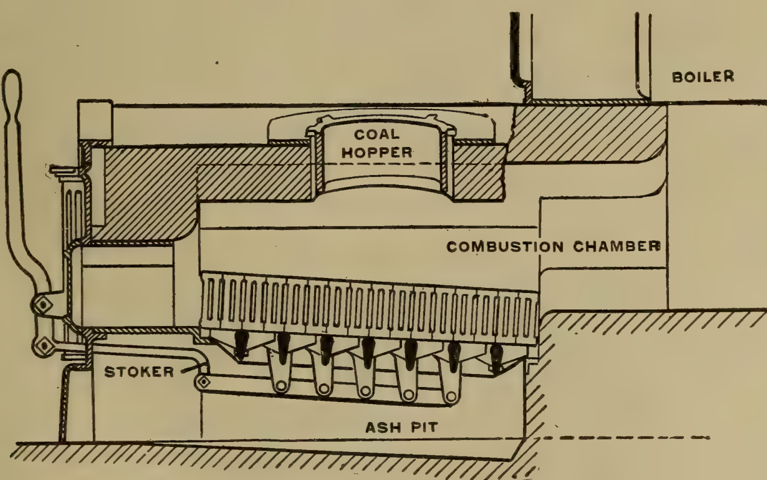


FIG. 91. Burke's Smokeless Furnace — Side Section.

to the upper grate, the lower one, formed of common bars, being fed by the half-consumed fuel falling from the upper grate. Air for combustion enters the upper fire door, which is kept open, and passes first through the bed of green fuel on the upper grate and then over the incandescent fuel on the lower grate. A strong draft is required, due

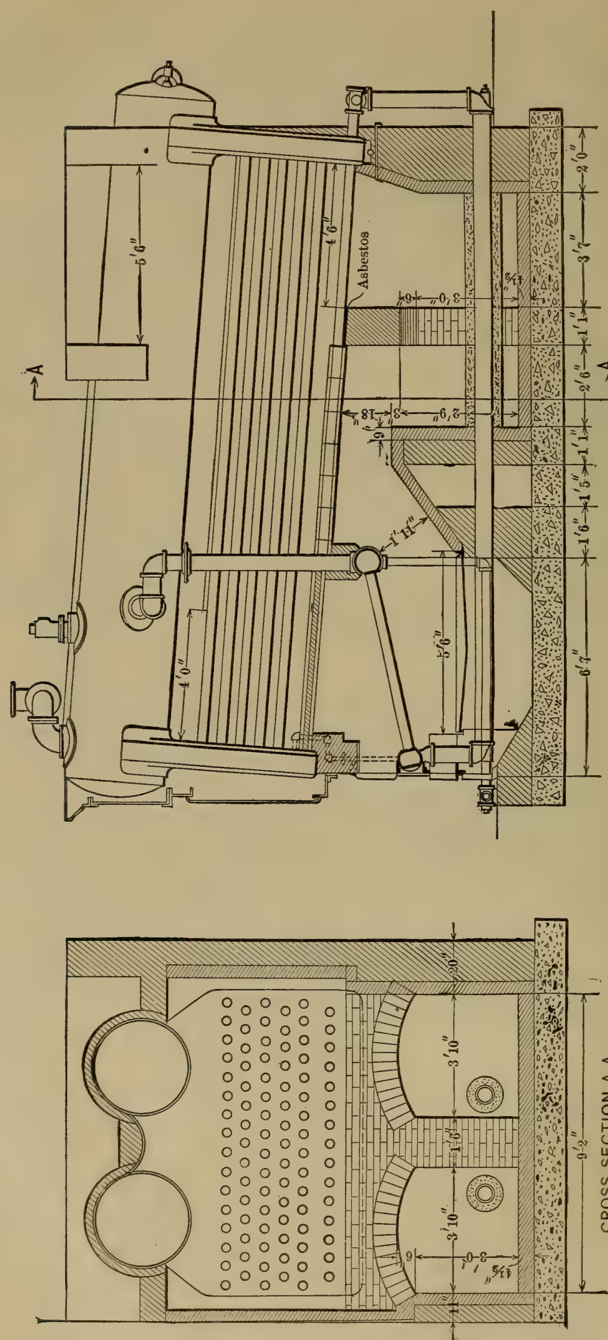


Fig. 92. Hawley Down-draft Furnace Applied to a Heine Boiler. (Chicago Setting.)

to the relatively small upper grate area and the correspondingly high rate of combustion. Lump coal gives better results than the smaller sizes, as the latter are apt to fall through the upper grate before being even partially consumed and when such is the case efficient results cannot be obtained. If carefully manipulated this furnace with fire-tiled tubes as illustrated in Fig. 92 gives high boiler efficiency and smokeless combustion, but its overload capacity is limited. Without the fire tiling smokeless combustion is possible only at light loads.

102. Steam Jets. — The main purpose of a steam jet in connection with "smokeless furnaces" is to mix the air and gases and insure intimate mixture of the products of combustion. This action is purely mechanical, the steam in itself not being a supporter of combustion. The claims sometimes made that steam increases the calorific value of fuel are erroneous. There are conditions with certain grades of coals and refuse under which a moderate amount of steam injected into the furnace promotes complete combustion and increases the efficiency of the boiler. Such results, however, are due to increase in *available* heat and not to increase in actual calorific value. For example, hydrogen and CO formed by the reaction between the steam and incandescent carbon unite with the oxygen of the air passing through the grate and generate intense heat. This heat dissociates a part of the steam into hydrogen and oxygen. The hydrogen immediately recombines with oxygen of the air, while the oxygen in its nascent state effects complete combustion of the hydrocarbons which under ordinary conditions escape in the form of smoke. Although it takes as much heat to dissociate steam into its elements as is given off when the hydrogen burns back again to water vapor the gain in available heat effected by the steam lies in the combustion of the hydrocarbons which would otherwise be discharged up the stack. The heat necessary to superheat the steam to stack temperature must be charged against the coal pile but the loss may be more than offset by this increase in available heat. It takes the same amount of oxygen to burn the hydrogen as is liberated by dissociation so there is no *extra* oxygen available for combustion, but the oxygen thus liberated is in a nascent state and combines much more readily with the hydrocarbons than does atmospheric oxygen.

There is no question as to the value of properly installed steam jets in maintaining smokeless combustion in internally fired furnaces, hand-fired return-tubular boilers and improperly designed furnaces, but taking all things into consideration better results may be had with properly designed furnaces equipped with mechanical stokers. A smokeless stack with hand firing is not a true indication of efficient operation, since the air dilution may be excessive and the heat demands

of the steam jets may be very great. Since air requirements are greatest at the moment of firing fresh coal, and the demand diminishes as distillation of the volatile matter progresses, steam jets need close regulation for best economy. If permitted to run continuously, as is often the case, they may use considerably more of the energy of the coal than they save by effecting smokeless combustion. In many of the patented "smoke consumers" the jets are automatic and operate independently of the fireman.

Fig. 93 illustrates the application of a steam jet to a hollow bridge wall. The top of the wall is fitted with a small cast-iron column *M*, partially imbedded in the brickwork. A series of 1-inch holes "*OO*," drilled near the top of the casting, furnish exits for the steam and air. A steam jet in one end of the column induces air into the iron chamber and forces it across the fire in fine streams. Excessive air dilution is

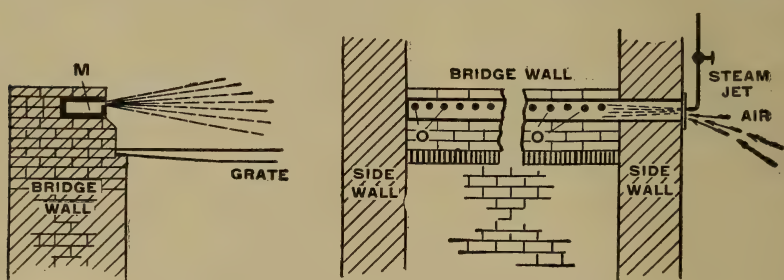


FIG. 93. Applications of Steam Jets to Hollow Bridge Wall.

avoided by partially closing the ash-pit doors and by regulating the intensity of the jets. An installation of this type is especially effective in connection with coal having a tendency to fuse and seal the air passages in the grate. Two Stirling boilers at the Armour Institute of Technology equipped with this device gave practically smokeless combustion at all normal loads, though at heavy overloads it was necessary to slightly open the fire doors. Without the use of the jets smoke could not be prevented except at light loads.

103. Parson Smokeless Furnace. — The Parson forced-draft system for smokeless combustion, applied to a return tubular boiler as illustrated in Fig. 94, comprises a specially designed grate *G*, depending upon a steam jet blower *A* for draft. Part of the steam is admitted below the grate and part over the fire through the hollow bridge wall *H*. The supply of air above the grate is regulated by means of damper *F*. The steam to blower *A* is automatically adjusted by regulator *N*, which is actuated by the steam pressure. The steam to the jet is superheated by passing the supply pipe through the setting as indicated.

The bridge wall H is provided with an extension platform M for holding the unburned fuel when cleaning the fire. When equipped with a properly designed deflecting arch the furnace meets with the requirements of the Chicago Smoke Department.

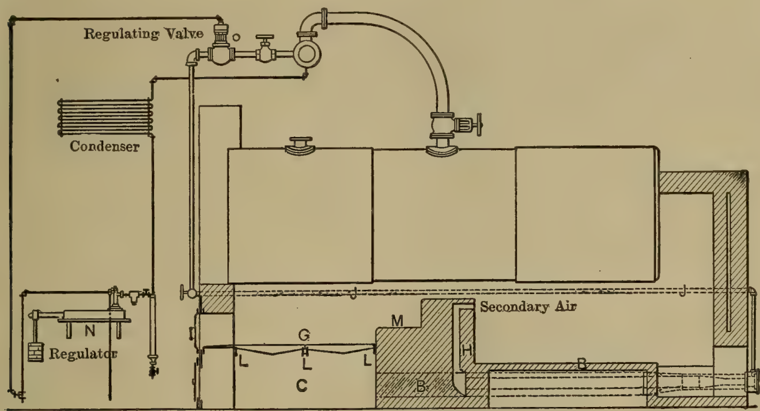


FIG. 94. Parson Smokeless Furnace.

104. Heinrich Smokeless Furnace. — Fig. 95 shows the application of the Heinrich system of forced draft to a return tubular boiler. Hot air is taken from the boiler room above the boilers by a steam jet blower at A and forced into the superheating chamber below the combustion chamber. From this chamber part of the air is drawn by the auxiliary blowers C and forced through tuyeres above the grate, the rest passing through an opening beneath the bridge wall into the ash pit and up through the bed of fuel. Steam for the blower A and the auxiliaries C is supplied through an automatic regulator R, which opens when the steam pressure falls below the required value.

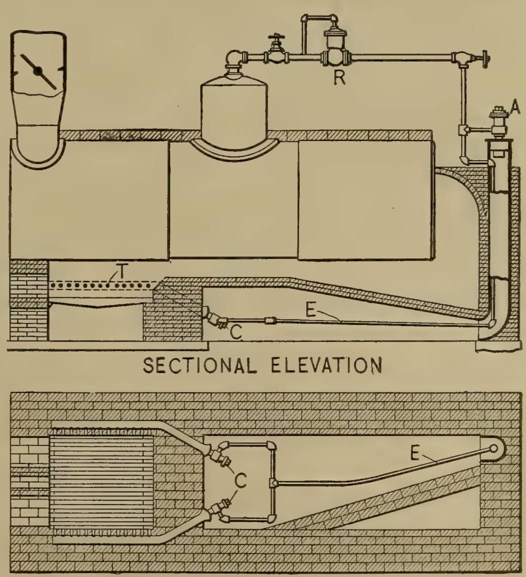


FIG. 95. Heinrich Smokeless Furnace.

105. Luckenbach Smokeless Furnace. — Fig. 96 shows the application of the Luckenbach system to a standard return tubular boiler and setting. It consists primarily of a set of needle nozzles placed in the side walls above the grate and supplied with highly superheated steam from a specially constructed superheater. The steam jets are automatically controlled by the fire door and in such a manner that opening the door opens the jets and closing the door cuts off the steam supply after a predetermined period. The superheater consists essentially of a wrought-iron coil cast within a solid block of iron and is placed on top of the bridge wall as indicated in the illustration. This superheater is an important feature of the system. It is absolutely foolproof and does not burn out even if the steam supply is entirely cut off. (For further information see Eng. News, Dec. 29, 1910.)

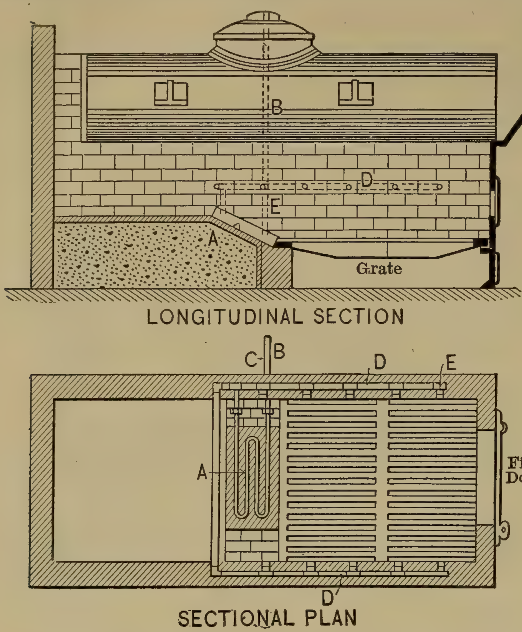


Fig. 96. Luckenbach Smokeless Furnace.

106. Mechanical Stokers. — *Uniform evolution of the volatile gases* of the fuel is the essential requisite for smokeless combustion, and it is for this reason that mechanical stokers as a class are more effective in preventing smoke than any apparatus accompanied by intermittent firing. Stokers which feed irregularly have the effect of hand-fired furnaces, and it is necessary not only to employ some powerful auxiliary mixing device but also to furnish at times an extra supply of air to take care of the enormous volume of volatile gas evolved after a fresh charge of fuel is added.

Carefully adjusted automatic stokers owe their high efficiency to: (1) *uniformity of feed;* (2) *proper proportion of air and combustible;* (3) *absence of excessive air dilution, as when the fire doors are opened in connection with hand firing;* and (4) *self-cleaning grates.*

Daily records are essential with any type of stoker or hand firing if efficient results are expected, as only by frequent observation is it

possible to determine the proper adjustment of air supply, depth of fire, rate of feed, and the like. Control of air supply is almost as important as the upkeep and effective operation. In the best firing practice the right amount of air, depth of fire, and rate of feed must be worked out by the engineer.

Stokers are often condemned by owners as inefficient and inferior to hand stoking because no particular attention has been paid to them beyond filling the hopper with coal. They should be operated in strict accordance with the principles of design.

In plants of 2000 horse power or over, the installation of mechanical stokers and coal conveyors effects a considerable saving of labor and can usually be relied upon to solve the smoke problem if reasonable attention is given to their operation. In smaller plants interest on the investment and other considerations may make hand firing more economical, although many plants of capacities as small as 200 horse power are giving satisfaction, particularly in places where a poor grade of fuel is used and smoke ordinances are rigidly enforced. A stoker of the self-cleaning, slow-running type requires much less attention than the hand-fired furnace. With hand firing one fireman can efficiently attend to the water, coal, and ashes of about 200 horse power, or handle coal for, say, 500 horse power, whereas with good automatic stokers he can readily take care of 2000 horse power or of 4000 horse power with chain grates equipped with overhead bunkers and down spouts.

The best stokers are those which are least complicated and simplest in operation. A cheap stoker is a poor investment, since the cost of repairs and shutdowns will usually amount to more than the saving in price.

The following outline gives a classification of a few of the best-known American mechanical stokers:

Front Feed.
- Chain Grates:
  - Babcock & Wilcox,
  - Green,
  - McKenzie,
  - Playford.
- Step Grates:
  - Roney,
  - Wilkinson,
  - Acme,
  - McClave.

Side Feed.
- Step Grates:
  - Murphy,
  - Detroit,
  - Model.

Under Feed.
- Jones,
- American,
- Taylor,
- Guckett.

Down Draft.
Hawley.

Powdered Fuel.
See paragraphs 34–46.

Sprinkler.
Swift,
Vulcan,
Erie.

107. Chain Grates. — The chain grate is one of the most popular forms of automatic stokers for burning small sizes of free-burning bituminous coals. It embodies a moving endless chain of grate bars mounted on a frame with provision for the continuous and uniform feeding of coal into the furnace, the fuel and the grate moving together. As usually installed the surface of the grate is horizontal though in some designs it is given a slight incline toward the bridge wall. The operations of feeding the coal, carrying it through the progressive stages of combustion, removing the ashes and clinkers, and maintaining a clean grate and free air supply are automatic. The driving mechanism consists of a gear train actuated by ratchet and pawls, the arms carrying the latter being given a reciprocating motion by an eccentric mounted on a line shaft. The latter may be driven by any type of engine or motor and the speed of the grate regulated by varying the stroke of the arm carrying the pawls. Fuel is fed into a hopper placed at the front end of the furnace and the depth of the fuel regulated by a guillotine damper. The front part of the furnace is provided with a flat or slightly inclined ignition arch the function of which is obvious. The entire grate and driving mechanism are mounted on a permanent truck and may readily be removed from beneath the boiler. The thickness of the fire and the speed of the grate should be so regulated that when the fuel has reached the end of the grate it shall have been completely consumed and ashes only will be discharged into the pit. With chain-grate stokers there may be considerable leakage of air between the grate and bridge wall, through the coal in the hoppers, under the coal-gate and through the fire bed at the rear where it is mostly ashes unless care is used in regulating the depth of fire and ash bed and provision is made for preventing this "short circuiting" of the air supply.

Fig. 97 shows the general application of a B. & W. chain grate to a B. & W. boiler. The ignition arch is parallel to the grate and covers a considerable portion of the grate surface. The bridge wall is fitted with a water back as indicated, to prevent the grate bars from being burned. With uniform loads this style of ignition arch and setting insures practically smokeless combustion, but careful manipulation is necessary with rapidly fluctuating loads to prevent the formation of objectionable smoke.

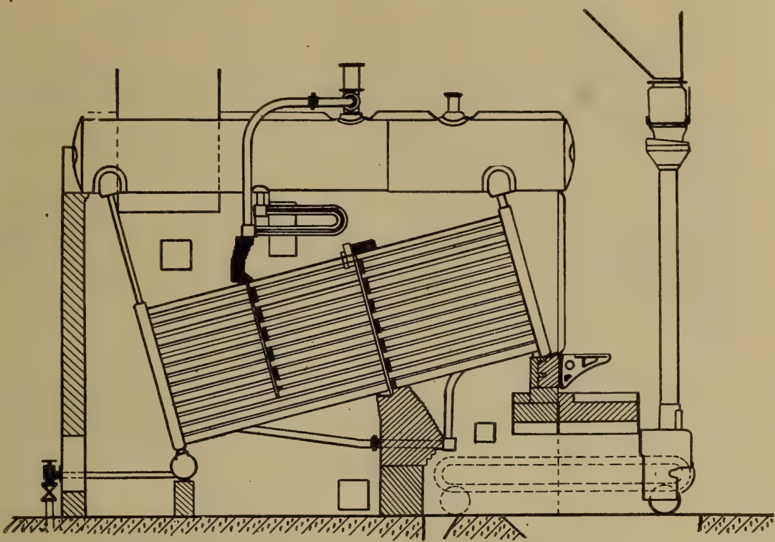


Fig. 97. Babcock and Wilcox Boiler, Chain Grate, Ordinary Setting.

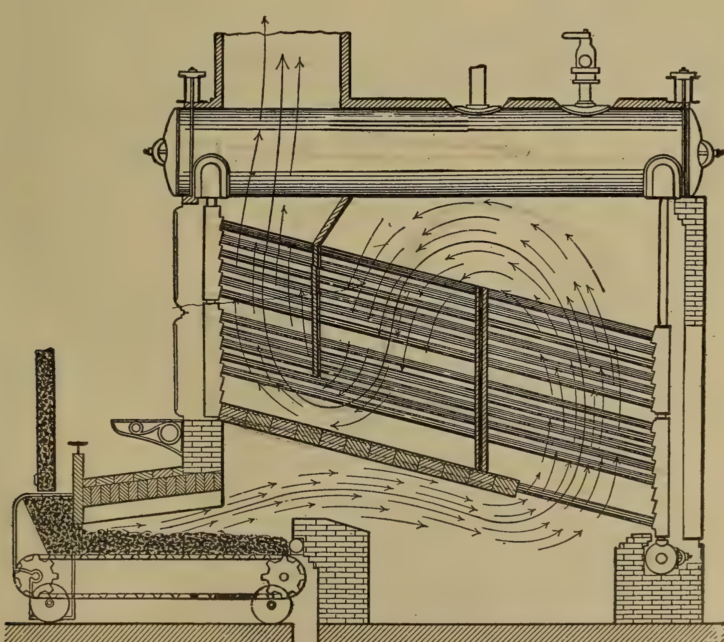


Fig. 98. Babcock and Wilcox Boiler, Chain Grate, Fire-tile Roof.

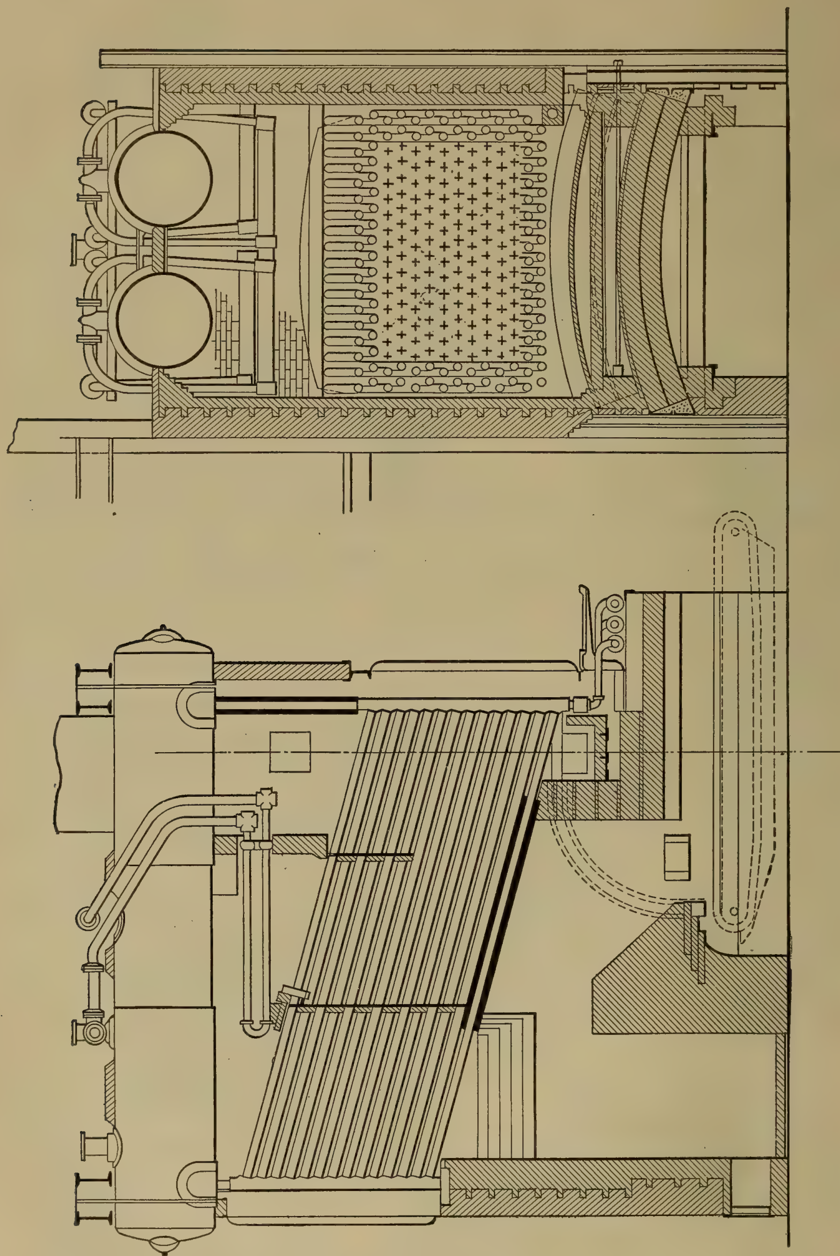


FIG. 99. Chain Grate Fired from Rear End of Setting as Installed at Quarry Street Station, Commonwealth Edison Co., Chicago.

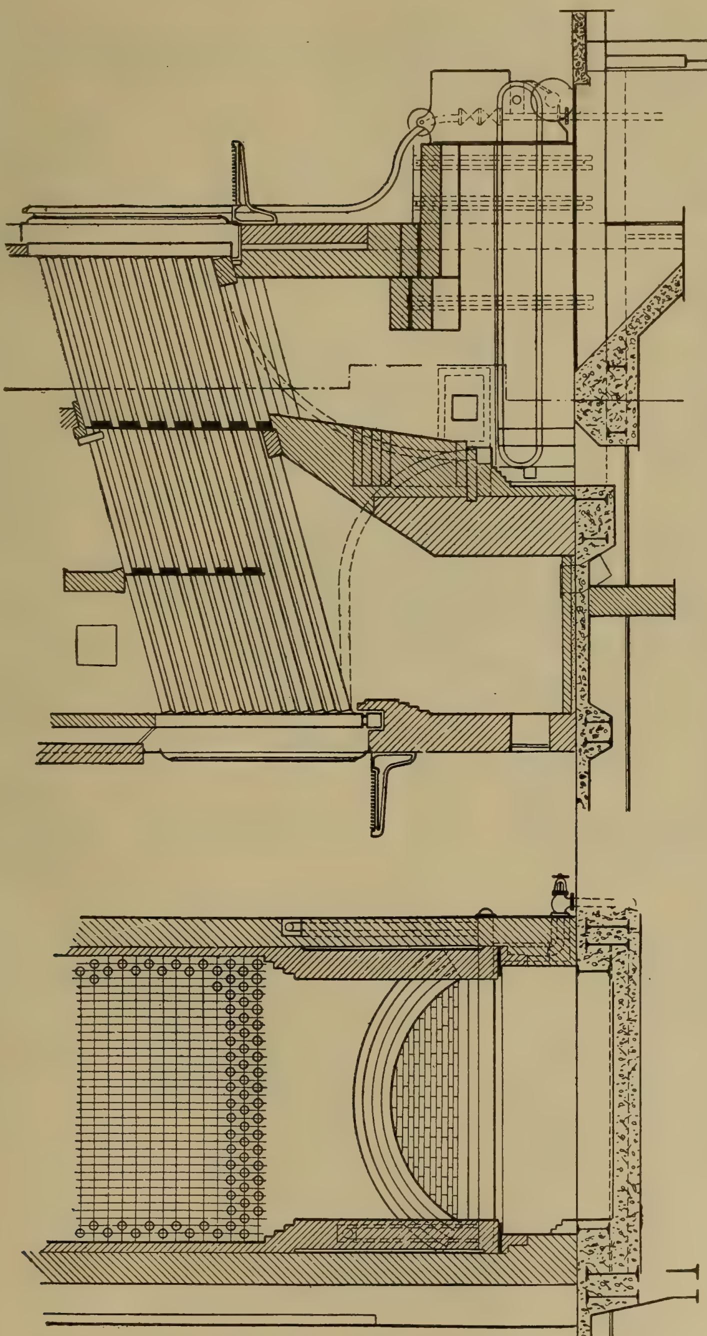


FIG. 100. Smokeless Setting, Boiler Units 5 and 6, Quarry Street Station, Commonwealth Edison Co., Chicago.

Fig. 98 shows the application of a fire-tile roof to a B. & W. boiler with chain grate. This arrangement, though effective in the prevention of smoke, is not much in evidence in modern plants because of the short life of the tiles.

Fig. 99 shows the application of a B. & W. chain grate to the rear of a B. & W. boiler as installed at the Quarry Street Station of the Commonwealth Edison Company. Fig. 100 shows the application of a B. & W. stoker to the front end of the setting but with a much larger combustion chamber. Both of these settings are effecting smokeless combustion with Illinois screenings.

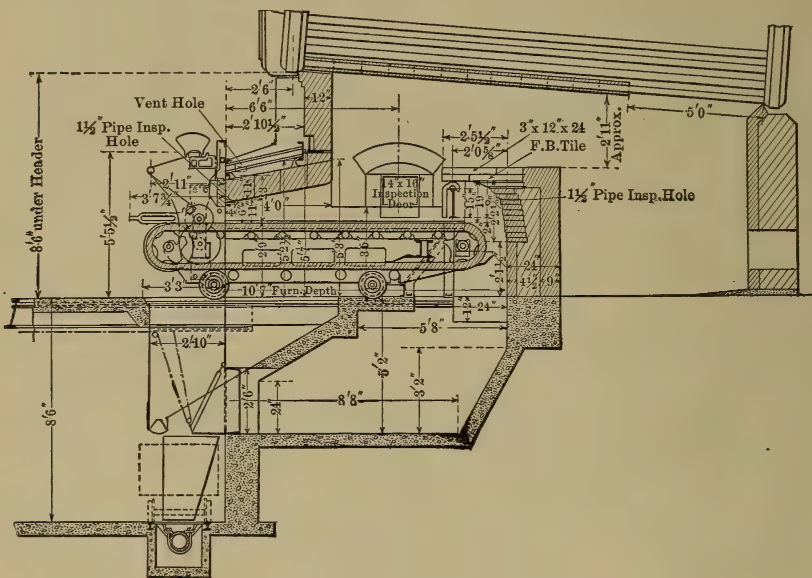


FIG. 101. Green Chain Grate, Standard Type.

Fig. 101 shows the application of a standard Green chain grate for free-burning bituminous coals. The ignition arch is inclined to the grate and the distance between arch and grate is much greater than with either of the settings just described. The arch is inclined so that the heat from the incandescent fuel bed near the bridge wall is reflected down upon the green fuel, thus hastening distillation of the volatile gases. This arrangement permits of greater overloads with smokeless combustion. Swinging "air-back" *E* serves the double purpose of retaining the fuel and ash on the grate until all of the combustible is consumed and of preventing air leakage between the bridge wall and the end of the chain. The accumulated ash is dumped into the ash pit as occasion requires by rotating the air-back on its axis. A small steam jet or air blast prevents the air-back from being overheated.

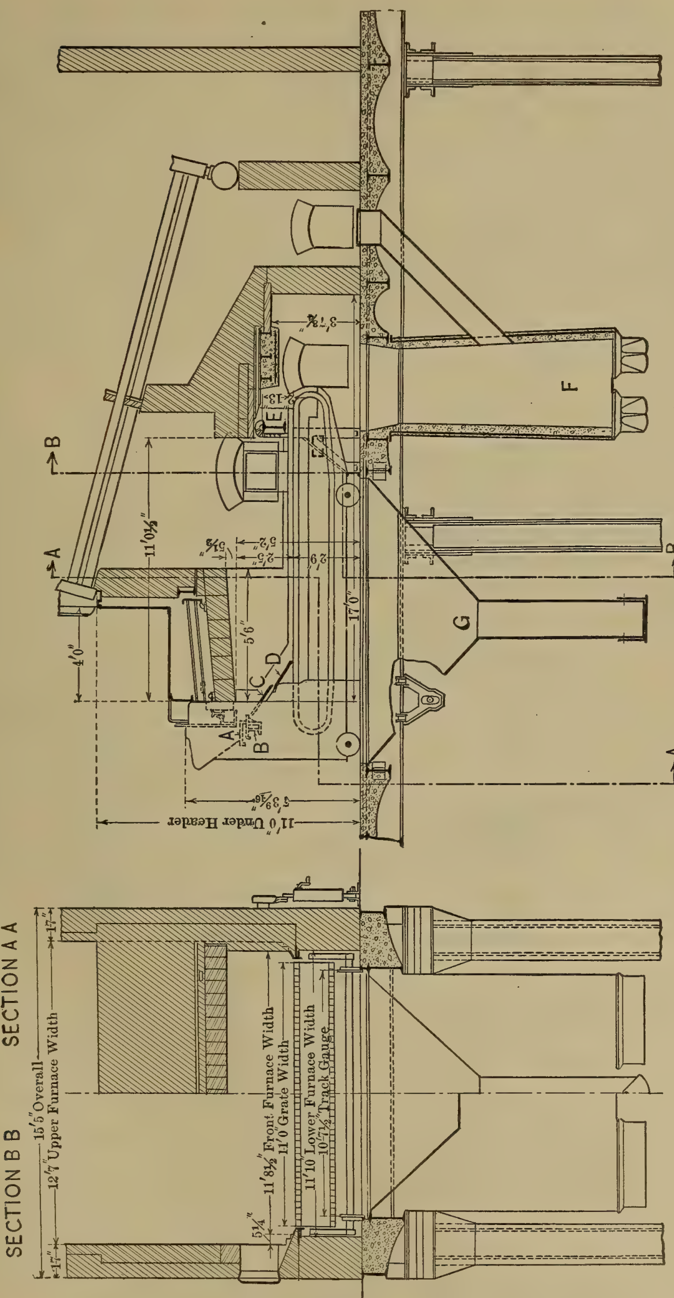
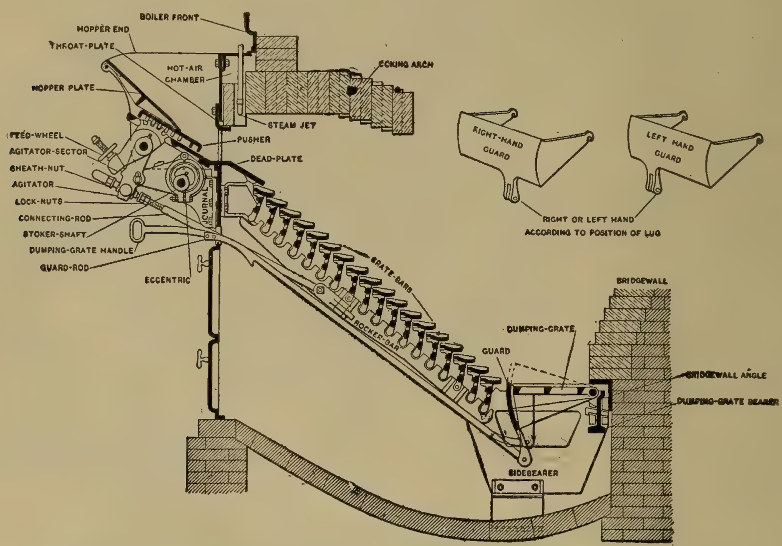


Fig. 102. Green Chain Grate, Type L.

Coking bituminous coals are not adaptable to the ordinary style of chain grate on account of the swelling and fusing action of the coal under the ignition arch. Fig. 102 shows a Green grate modified to burn this grade of fuel. The fresh fuel is fed from the hopper onto dead plate *B* by means of pusher *A* and from the latter upon the inclined coking plates, *C*, *D*. These plates gently agitate the fuel during the period of distillation and prevent it from fusing together so that by the time it reaches the grate proper, it no longer tends to cake. The ignition arch is inclined for the purpose previously stated. Grates of this type installed at the 59th Street station of the Interborough Rapid Transit Company are giving excellent results at heavy overloads. The results of these tests are shown in Fig. 57.



Details of Construction of the Roney Mechanical Stoker

Fig. 103. Details of Roney Stoker.

108. Step Grates, Front Feed. — Fig. 103 shows the general arrangement of a Roney stoker and Fig. 105 that of a Wilkinson stoker, illustrating the step-grate, front-feed principle. The Roney stoker consists of a hopper for receiving the coal, a set of rocking stepped grates inclined at a proper angle from the horizontal, and a dumping grate at the bottom of the incline for receiving and discharging the ash and clinkers. The dumping grate is divided into several sections for convenience in handling. The coal is fed onto the inclined grate from the hopper by a reciprocating "pusher" actuated by the "agitator." The power is supplied through an eccentric operated by a small engine or motor. The normal feed is about 10 strokes per minute. The grate bars rock

through an arc of 30 degrees, assuming alternately horizontal and inclined positions. The construction permits abundance of air to pass through the fuel, with little or no possibility of coal dropping through the grate. A coking arch of fire brick is sprung across the furnace as indicated. This stoker operates with natural draft and, with suitable

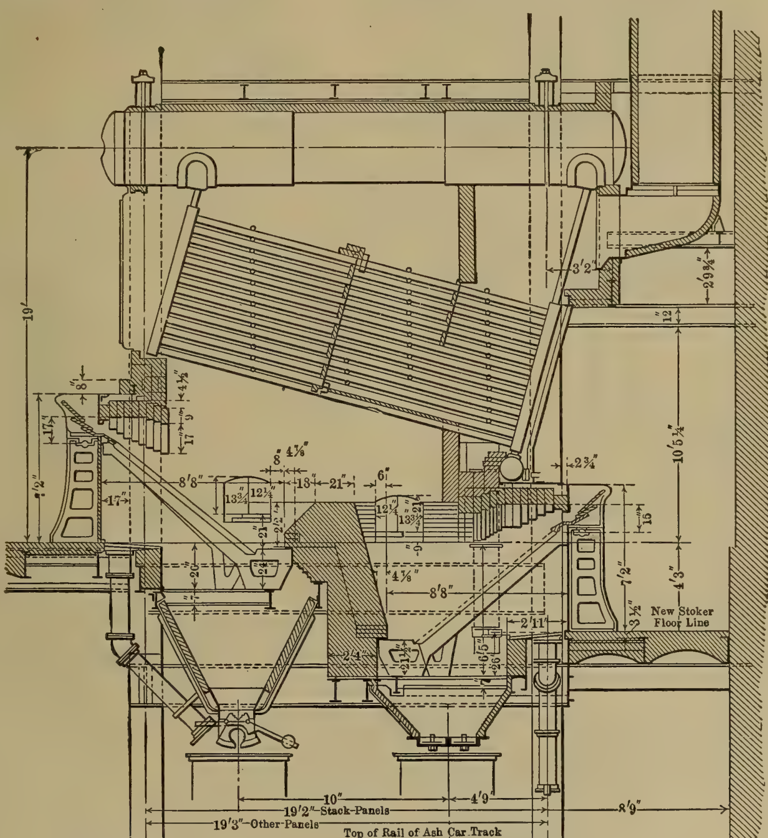


FIG. 104. Double Stoker Installation at the 59th Street Station of the Interborough Rapid Transit Co., N. Y.

arrangement of fire tiling, effects complete and efficient combustion. Without a fire-tile roof construction, smokeless combustion is effected with difficulty, particularly at heavy loads.

In the Wilkinson stoker the inclined grate bars are hollow and are arranged side by side, every alternate bar being movable. When in operation there is a constant sawing action of the grate bars, causing the fuel to flow forward and downward. A small steam jet with about $\frac{1}{16}$-inch opening is introduced into the end of each hollow grate bar,

thus inducing the required amount of air for combustion, which passes through air openings approximately $\frac{1}{4}$-inch wide by 3 inches long. These stokers are driven by two small hydraulic motors, the water being furnished by a small pump and being used over and over again.

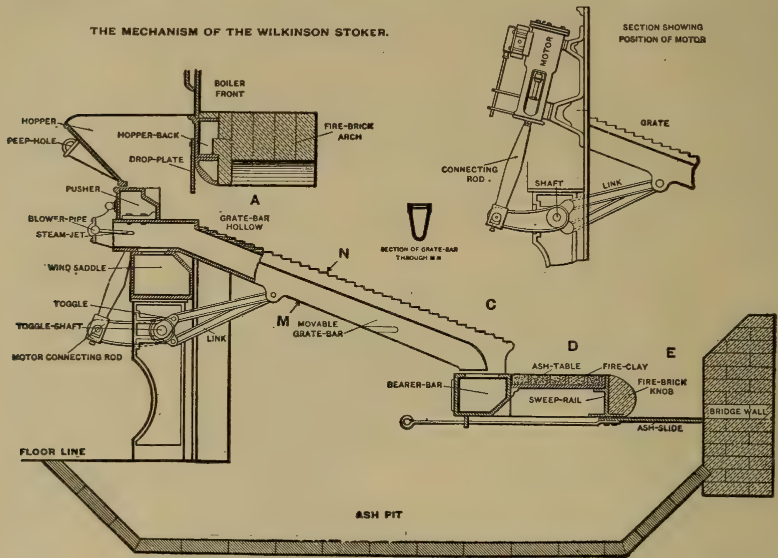


FIG. 105. Details of Wilkinson Stoker.

109. Step Grates, Side Feed. — Fig. 106 shows a front vertical section and Fig. 107 a side vertical section through a Murphy automatic stoker and furnace. The apparatus is in effect a Dutch oven equipped with an automatic feeding and stoking device. Coal is introduced either mechanically or by hand into the magazine at each side of the furnace and above the grate and descends by gravity upon the coking plate. Reciprocating stoker boxes push the coal out upon the grate bars. Every alternate grate bar is movable and pivoted at its upper end. A rocker bar driven by a small motor or engine causes the lower ends to move up and down, this action producing the required stoking effect. A device for grinding up the clinker and ash is provided as shown at the bottom of the furnace. This is hollow and is connected by a 2-inch pipe with the smoke flue, so that the cold air passing through it prevents it being destroyed by the heat. Air is supplied to the green coal through flues passing under the coking plates, and the speed of the stoker boxes and grate bars can be regulated to conform to any rate of combustion. On account of the large fire-brick combustion chamber, this stoker with *careful manipulation* is capable of practically smokeless combustion. The power house of the Northwestern Elevated

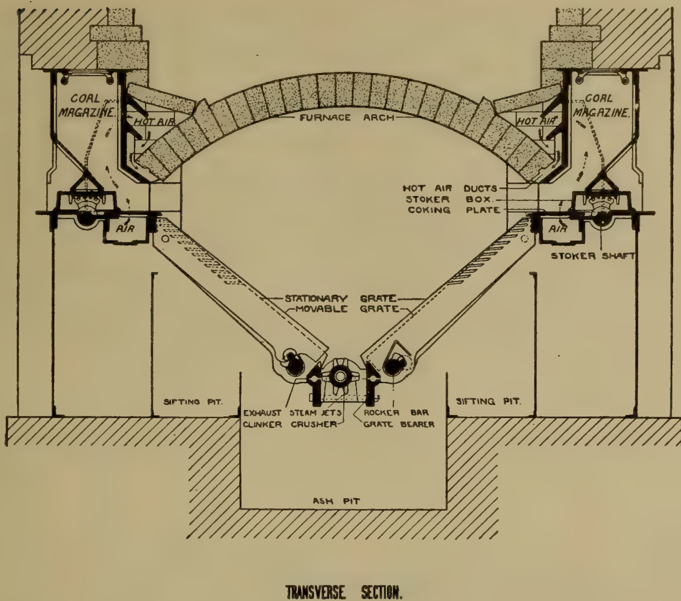


FIG. 106. Murphy Furnace, Front Section.

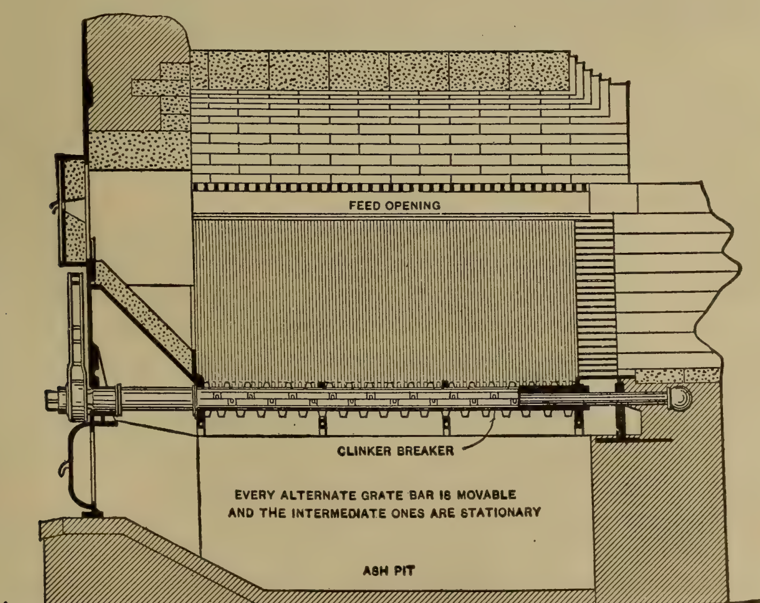


FIG. 107. Murphy Furnace, Side Section.

Railroad Company, Chicago, Ill., is equipped with Murphy furnaces, which are operating smokelessly at an unusually high combustion rate, whereas a number of other installations using the same type of stoker and boiler and burning the same class of fuel are heavy smoke producers. Murphy furnaces are peculiarly adapted to variable loads, since at light loads the stoker may be operated with reduced grate area by allowing the bottom of the grate to partly fill with ashes.

110. Jones Underfeed Stoker. — Fig. 108 shows the general principles of the Jones underfeed stoker. It consists of a steam-actuated ram with a fuel hopper outside of the furnace proper and a fuel magazine and auxiliary ram within. Air for combustion is admitted through openings in the tuyere blocks on either side of the retort. Coal is fed into hoppers and forced *under* the bed of fuel in the stoker retort, where it is subjected to a coking action. After liberation of the volatile gases the coke is pushed toward the top of the fire. The top of the

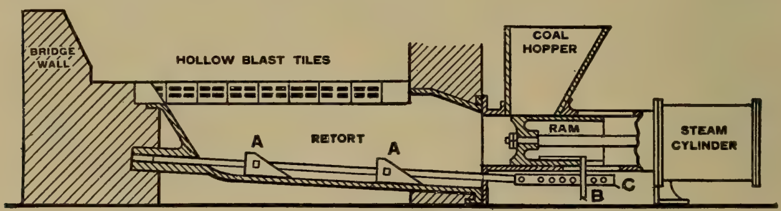


FIG. 108. Jones Underfeed Stoker.

fire, nearest the boiler, is always incandescent. Each charge of coal gives an upward and backward movement, which agitates the fire without opening fire doors. Air is admitted through the tuyere blocks at the point of distillation of the gases.

Grate bars form no part of the Jones system, and it is therefore impossible for the fuel to fall through. There is no ash pit. The noncombustible matter is removed from the furnace by hand. The standard size of the retort is about 6 feet in length, 28 inches in width, and 18 inches in depth, and experience has shown that other sizes are not necessary since the spaces between retort and side wall of the various furnaces may be provided for by extending the width of the dead plates. One or more stokers are installed in each furnace, depending upon the capacity of the boiler and the width of the furnace.

The steam pressure automatically controls air and fuel supply, proportioning them to each other and to varying loads in the correct degree. The result is that the stoker effects complete and smokeless combustion. The only variable element in the operation of this stoker, once it is correctly installed, is cleaning of fires, but if the fireman is

careful to burn down the coals before breaking them up the production of smoke may be avoided.

Jones underfeed stokers are adaptable to all grades and sizes of bituminous coal, and on account of forced draft are capable of burning very low grades of coal.

111. American Underfeed Stoker. — Fig. 109 shows an application of an American underfeed stoker to a return tubular boiler. This differs from the Jones stoker in the method of feeding the fuel to the retort and in the employment of "live" grates instead of dead plates on the sides of the retort. The coal is fed into the hopper and carried by

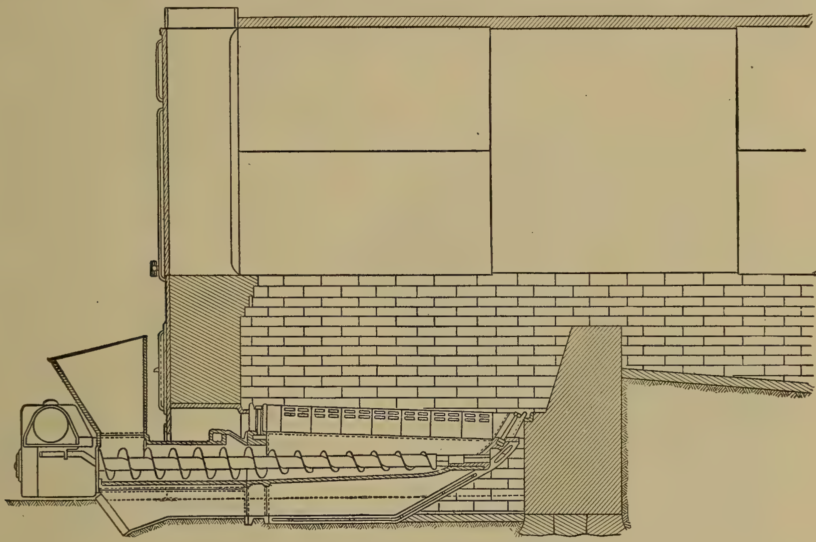


Fig. 109. American Underfeed Stoker.

an endless screw conveyor into the magazine or retort. Forced draft is used and the air supply and speed of the conveyor are readily adjusted to suit the conditions of load. In all underfeed stokers complete combustion is effected within a very short distance from the retort, hence a much smaller combustion space is required than with other types of stokers. For this reason underfeed stokers may give good results when installed in internally-fired boilers and small hand-fired return tubular boilers.

112. Taylor Underfeed Stoker. — Fig. 110 shows the general details of a Taylor underfeed stoker for burning bituminous coals. The device consists essentially of a series of alternate retorts and tuyere boxes inclined as indicated. Each retort is fitted with two rams — the upper for pushing the green fuel outward and upward and the lower one for forcing the fuel bed and refuse toward the dump plates at the

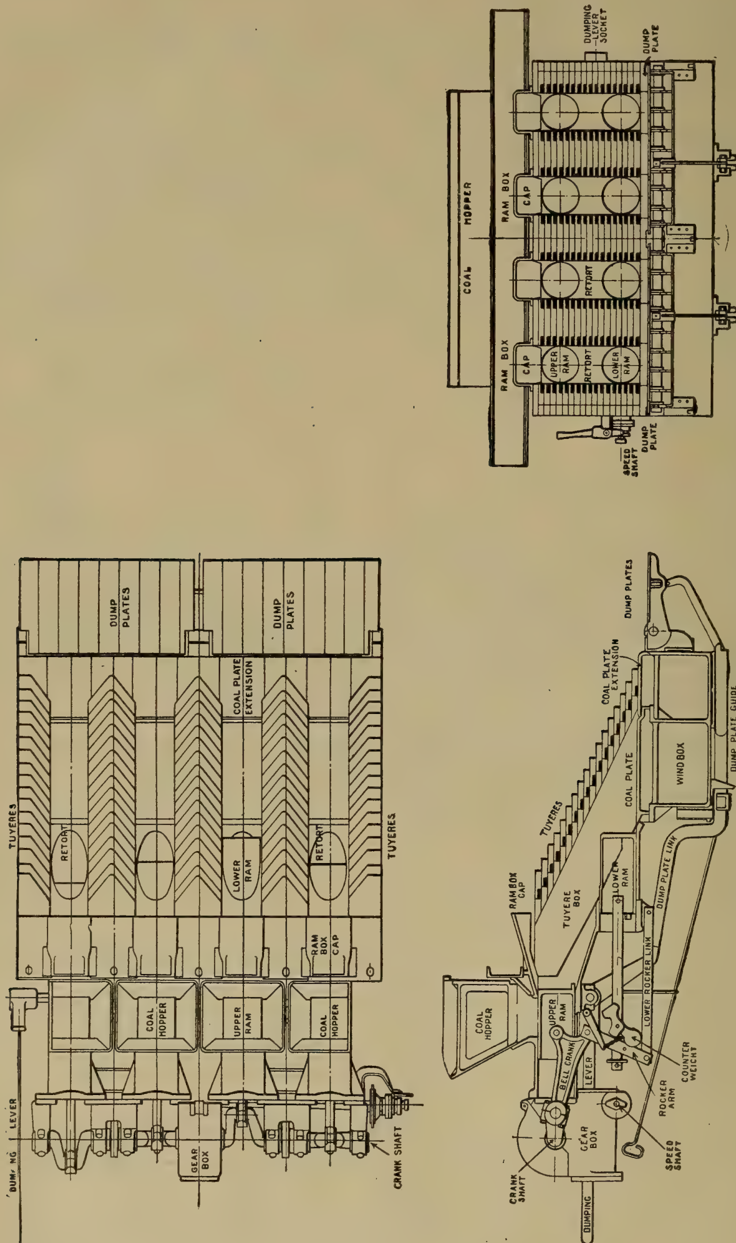


FIG. 110. General Details of the Taylor Underfeed Stoker.

rear. Air is supplied by a volume blower and enters the furnace through openings in the tuyere boxes. The dump plates are hung on the rear of the wind box and are controlled from the front of the stoker. Extension grates are inserted between the mouth of the retort and the dump plates, when the nature of the fuel makes this arrangement desirable. This extension may be rocked if necessary. The stoker and blower are operated by the same engine, the air and coal supply being automatically controlled by the variation in steam pressure. Taylor stokers may be operated smokelessly and efficiently at very heavy overloads and are much in evidence in the eastern states. The steam required to operate the blower and stoker varies from 2.5 to 5 per cent of the steam generated, depending upon the size of the installation and the percentage of rating developed.

113. Sprinkling Stokers. — In this system of stoking the fuel in finely divided form is distributed by sprinkling uniformly over the entire area of the grate. With the proper adjustment of air supply and feed the volatile gases are distilled off continuously before the grate is covered by the new coal and without materially lowering the temperature of the incandescent fuel. Mechanically the operation involves considerable difficulty. Sprinkling stokers do not conform to Chicago requirements.

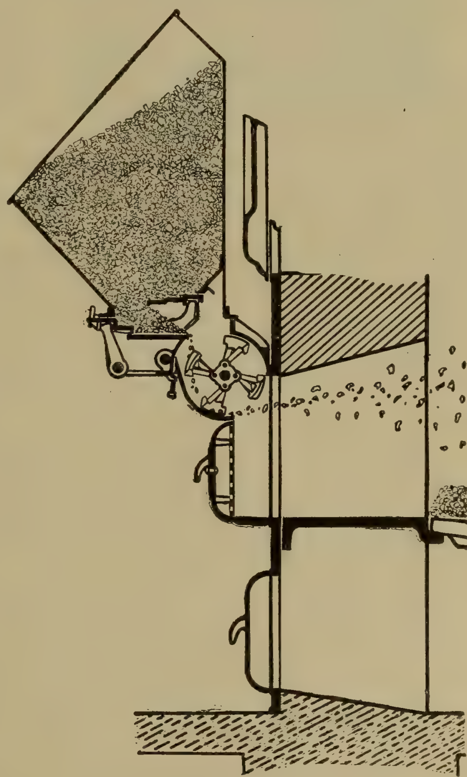


FIG. 111. Swift Sprinkling Stoker.

Fig. 111 gives the general details of the Swift stoker, illustrating a commercially successful stoker of this type. The apparatus is self-contained, and is bolted to a frame casting in front of the setting, and takes the place of the fire door. It may be swung back from the fire-door opening in much the same manner as the ordinary fire door. Coal of nut size or smaller is fed into a small hopper, of about 300 pounds' capacity, from which it gravitates on to a berm plate and

pusher plate. By means of the latter the fuel is fed to rapidly revolving spreaders, which crush it into small particles and throw it onto the grate. The fine or powdered coal is burned in suspension and the heavier coal falls to the grate. The spreaders are heavy pieces of cast steel, revolving about a common axis and shaped helically so as to throw the fuel in a direction at right angles to the face of the machine. There are several of these spreaders so arranged on the shaft that adjacent spreaders throw the fuel in different directions. This stoker is not self-cleansing, that is, the ashes must be removed by hand or by suitable shaking grates.

114. Smoke Determinations. — Smoke measurements may be either quantitative or relative.

The most satisfactory method, at this writing, of determining the quantity of smoke passing through a chimney is that adopted by the Chicago Association of Commerce. A continuous sample of chimney gas is drawn from the stack by means of a special Pitot tube and exhauster, and the solid particles are entrapped in a filter. The tube is so arranged that the rate of flow through the apparatus is the same as that in the chimney. Since the area of the tube opening bears a fixed ratio to that of the chimney, the weight of carbon, cinders, soot and the like caught in the tube filter is a measure of the total weight emitted from the stack.

Quantitative measurements are of considerable value in estimating the amount of energy lost in the production of visible smoke, but are seldom attempted in regular practice.

There are several methods of determining smoke, relatively. The most common is that devised by Ringelmann, and is commercially known as the Ringelmann Smoke Chart. The chart, as published by the U. S. Geological Survey and used by the Smoke Department of the City of Chicago and other municipalities, consists essentially of a cardboard folder 12 by 26 inches over all. Four charts are printed on this folder, each chart consisting of 294 squares, 14 squares wide by 21 squares in length, the width of the lines and spacings varying as illustrated in Fig. 112. At a distance of 50 feet from the observer the lines become invisible and the cards appear to be of different shades of gray, ranging from very light gray to almost black. The observer places the chart on a level with the eye (at the distance stated, and as nearly as possible in line with the chimney) and notes which card most nearly corresponds with the color of the smoke. Observations should be made at 15-second intervals and recorded as in Fig. 113. No smoke is recorded as No. 0, 100 per cent as No. 5, and the intermediate colors as indicated by the cards.

Experienced observers often record in half-chart numbers. Although these observations depend upon the personal element it is the opinion of the Chicago Smoke Department that only a little experience is necessary to effect consistent results with different observers.

Prior to 1910 a chimney was held to be a smoke nuisance by the Chicago smoke inspection authorities when it emitted smoke of No. 3

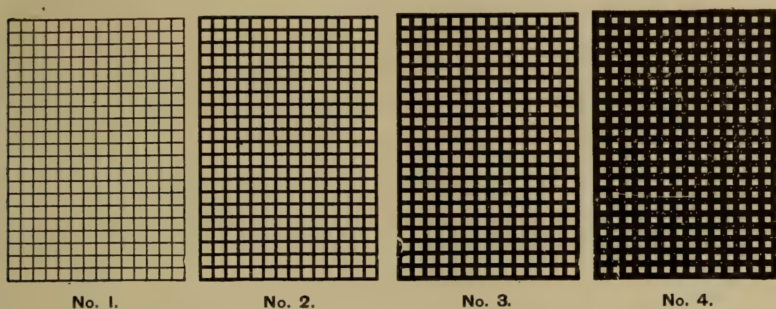


FIG. 112. Ringelmann Smoke Chart (Greatly Reduced).

density, according to the Ringelmann chart, for 7 minutes during one hour, as based on the original ordinance. With this standard the owners of a chimney which emitted but a very small total quantity of smoke might be liable to punishment, whereas, with a chimney which continuously emitted smoke of a density less than No. 3, the owners would be safe from legal prosecution, although the total quantity emitted might be many times as great.

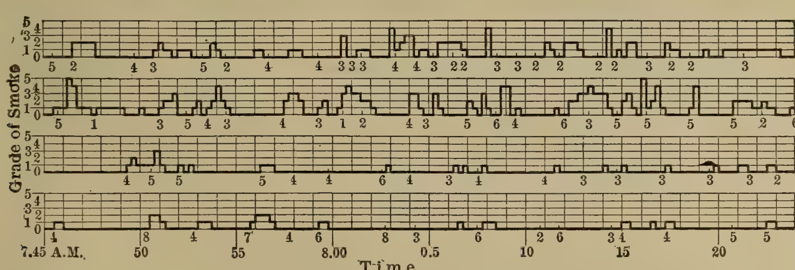


FIG. 113. Smoke Record Chart.

In future, the total smoke emitted will be taken into consideration. Observations will be made on a given stack every 15 seconds throughout the entire day and the total "smoke units" will be recorded, from which the average smoke density for the entire period will be calculated.

A "smoke unit" is the equivalent of No. 1 smoke (Ringelmann scale) emitted for one minute. No. 1 smoke has a density of 20 per

cent; No. 2, 40; No. 3, 60; No. 4, 80, and No. 5, 100 per cent. Thus, if a stack emits No. 3 smoke for 6 minutes, 18 smoke units are charged against it. If this smoke was emitted during one hour's observation, then

$$\frac{3 \times 6 \times 20}{60} = 6 \text{ per cent}$$

is the average density of smoke emitted during the period of observation.

If observations on a given stack show that the density averages more than 2 per cent, although the owner may not be legally liable, an appeal is made to his personal and civic pride by a representative of the smoke-inspection department. For example, if a certain hotel stack emits smoke of more than 2 per cent average density, the smoke department finds a plant record of similar design and equipment, preferably a hotel plant, which shows a record well below the 2 per cent mark. This plant is then pointed out to the owner or manager having the objectionable chimney and he is asked if he cannot do equally well when he has practically the same equipment, etc.

It has been found that this method of procedure often produces quicker and better results than a threat to go to law.

New Methods of Approaching the Smoke Problem, Osborn Monnett, Jour. Wes. Soc. Engrs., Nov. 4, 1912.

DIVISIONS OF MESH: RINGELMANN'S SMOKE CHART.

Numbers give Relative Smoke Density.	Thickness of Lines, mm.	Distance in the Clear between Lines, mm.
0	All white	All white
1	1	9.0
2	2.3	7.7
3	3.7	6.3
4	5.5	4.5
5	All black	....

The Hammler-Eddy smoke recorder, Fig. 114, is one of the most successful devices for automatically recording the density of the smoke independent of personal observations. This apparatus consists essentially of a small motor-driven vacuum pump, which draws a continuous sample of the products of combustion from the uptake, breeching or stack and discharges it against a paper-covered drum revolved by clockwork. The *density* of the smoke, the *time* at which visible smoke is being emitted and the *duration* of the smoke-production period are automatically recorded on the paper by the smoke itself. Before

reaching the pumps the gases pass through a glass "emergency" condenser and a large portion of the vapor content is removed. The pump discharges the partially dried gases against a surface of sulphuric acid (which removes the last trace of moisture) and forces the smoke in the form of a small jet of dry powder onto the surface of the recording paper. The sampling tube leading from the flue to the pump is

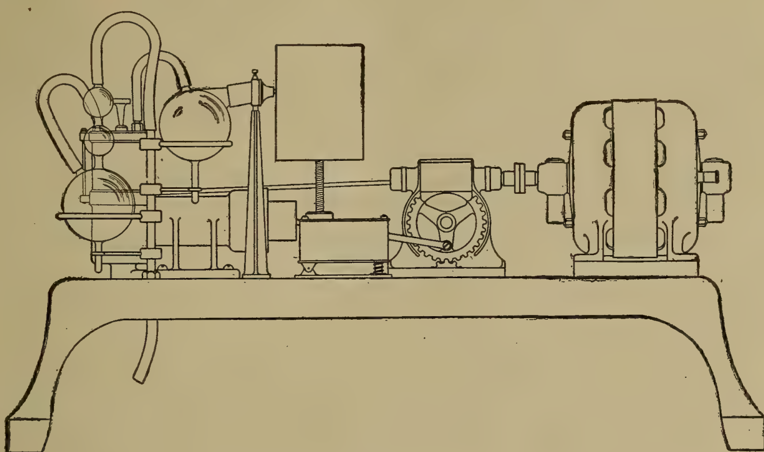


FIG. 114. Hammler-Eddy Smoke Recorder — Motor-driven Type.

connected with a steam line and is "blown out" each time a card is changed. The instrument is very compact and portable and may be placed anywhere with respect to the chimney. A number of these appliances in Chicago power plants are giving excellent satisfaction. In a more recent design the pump is replaced by a steam siphon.

115. Cost of Stokers. — The following is the approximate cost of stokers suitable for a Babcock and Wilcox boiler of 350-horse-power rated capacity with 45 square feet of grate surface; height of chimney above grate, 175 feet; coal burned, Illinois screenings. The cost of installation included, exclusive of brickwork, is

1. Chain grate and appurtenances	$1,500.00
2. Jones underfeed stoker	1,400.00
3. Hawley down-draft furnace	1,350.00
4. Burke smokeless furnace	1,000.00
5. Roney stoker	1,300.00
6. Murphy furnace and stoker	1,350.00
7. Wilkinson stoker	1,200.00

CHAPTER V.

SUPERHEATED STEAM; SUPERHEATERS.

116. General. — The steam engine fails to realize the efficiency of the ideal engine chiefly on account of cylinder condensation. The loss in heat due to this cause is seldom less than 10 per cent of the total supplied, and often as great as 40 per cent.

If the steam is superheated before being admitted to the cylinder, condensation may be reduced or prevented entirely, as was recognized as early as sixty years ago, but the mechanical difficulties encountered prevented the practice until within the past few years.

The principal advantages of superheated steam in connection with steam-engine work are:

1. At high temperatures it behaves like a gas and is, therefore, in a far more stable condition than in the saturated form. Considerable heat may be abstracted without producing liquefaction, whereas the slightest absorption of heat from saturated steam results in condensation. If superheat is high enough to supply not only the heat absorbed by the cylinder walls but also the heat equivalent of the work done during expansion, then the steam will be dry and saturated at release. This is the condition of maximum efficiency in a single cylinder. (Ripper, "Steam Engine Theory," p. 155.) Greater superheat than this will result in a loss of energy unless the steam is exhausted into another cylinder. To obtain dry steam at release the steam at cut-off must be superheated from 100 to 300 degrees F. above saturation temperature, depending upon the initial condition of the steam and the number of expansions, a higher degree of superheat being required for earlier cut-off. A superheat of from 250 to 350 degrees F. at admission is necessary to insure dry steam at release in the average single-cylinder engine cutting off at one-fourth stroke, boiler pressure 100 pounds gauge. In most cases superheat is only carried so far as to reduce initial condensation, the steam becoming saturated at cut-off, thus permitting efficient lubrication. There will be a reduction of approximately 1 per cent in cylinder condensation for every 7.5 to 10 degrees of superheat. In compound and triple-expansion engines the steam is ordinarily superheated between each stage as well as before admission to the high-pressure cylinder.

2. A moderate amount of superheat produces a large increase in volume, the pressure remaining constant, and diminishes the weight of steam per stroke for a given amount of work. For example, the volume of one pound of saturated steam at 165 pounds pressure (absolute) is 2.75 cubic feet, and its temperature is 366 degrees F. The total heat of one pound of this steam above the freezing point is 1195 B.t.u. By adding 108 B.t.u. in the form of superheat its temperature will be increased to 565.8 degrees F. (superheated 200 degrees F.) and its volume to 3.68 cubic feet (specific heat taken as 0.54). Thus an increase of 9 per cent in the heat effects an increase of 34 per cent in the volume, which means a corresponding reduction in the weight of steam admitted to the engine per stroke. These figures are purely theoretical, as no allowances have been made for condensation of the saturated steam or for reduction in temperature of the superheated steam.

3. Superheated steam has a much lower thermal conductivity than saturated steam, and, therefore, less heat is absorbed per unit of time by the cylinder walls. With superheat smaller steam pipes may be used or a greater amount of power transmitted in a pipe of given size. By using high pressure and high superheat and then by lowering the pressure with reducing valves at the end of the line it is possible to transmit steam 10,000 feet or more without serious heat losses.

117. Economy of Superheat. — Many comparative tests of engines using saturated and superheated steam under varying conditions of pressure and temperature have been made during the past few years, showing in all cases a gain in favor of superheat due to the reduction in steam consumption. In the majority of superheated steam installations the ultimate gain is a substantial one, but in some cases the extra investment and cost of maintenance neutralize the reduction in steam consumption, resulting in an actual loss when measured in dollars and cents per horse-power hour.

As far as steam consumption per horse-power hour is concerned, superheating usually increases the economy from 5 to 15 per cent and in some instances as much as 40, the latter figure referring to the more wasteful types of engines. A fair estimate of the average reduction in steam consumption per horse-power hour with moderate superheating, that is, from 100 to 125 degrees F., based on continuous operation of existing plants, is:

	Per Cent.
1. Slow running, full stroke, or throttling engines, including direct acting pumps	40
2. Simple engines, non-condensing, with medium piston speed, including compound direct acting pumps	20
3. Compound condensing Corliss engines	10
4. Triple-expansion engines	6

European builders guarantee steam consumption with highly superheated steam as follows:

	Pounds per i.h.p. hour.
Single-cylinder condensing engines (uniflow)	8.5
Single-cylinder non-condensing engines (uniflow)	12.0
Compound condensing engines (locomobile)	8.0
Compound non-condensing engines (locomobile)	10.5

The best recorded steam consumption at this writing (June, 1912) is that of a locomobile compound using steam superheated to 806 degrees F. at an initial pressure of 220 pounds absolute. When exhausting against an absolute back pressure of 1.32 pounds the steam consumption was 6.95 pounds per i.h.p. hour. (Zeit. des Ver. deut. Ingr., Mar. 18, 1911, p. 415.)

In comparing the performances of engines using saturated and superheated steam it is advisable to base all results on the *heat* consumed per horse power rather than on the steam consumption, since the latter is apt to give a false idea of the relative economies. The real measure of economy is the *cost* of producing power, taking into consideration all charges, fixed and operating, and the next best is the coal consumption per i.h.p. hour, but as a means of comparing the engines only, the heat consumption per horse power per hour or per minute is very satisfactory. (See paragraph 180.)

See paragraph 208 for the influence of superheat on the economy of reciprocating engines and paragraph 235 for the influence on steam turbines.

118. Limit of Superheat. — In this country steam temperatures exceeding 500 degrees F. are seldom employed, while in Europe few if any plants are installed without superheaters, and 600 degrees F. is a common temperature with a maximum of about 850 degrees F.

Experience has shown that with engines of ordinary design, slide-valves and Corliss, the temperature at the throttle should not exceed 500 degrees F. This corresponds to a superheat of 160 degrees F. with steam at 100 pounds gauge pressure, and 130 degrees F. at 150 pounds. This degree of superheat insures practically dry steam at cut-off in the better grade of engines. Just how far superheating can be carried with a given engine of ordinary construction can be determined by experiment only, but a temperature of 500 degrees F. is probably an outside figure and 450 degrees F. a good average. Higher temperatures are apt to interfere with lubrication and sometimes cause warping of the valves. With temperatures below 450 degrees F. no difficulties are ordinarily met with. Metallic packing has been found to give the best results for both piston rods and valve stems. For highly super-

heated steam "labyrinth" packing is used in place of the ordinary flexible metallic packing.

It is generally assumed that a greater quantity of oil is required for lubricating valves and cylinders in connection with superheated steam, but experience seems to show that such is not the case. (Proc. A.S.M.E., May 14, 1908.) Forced-feed lubricators are the most satisfactory for superheated steam engines, since they insure a positive and copious flow of oil directly to the valves or other parts requiring it. (Effect of Superheated Steam on Cylinder Oils. Mech. Engr., Lond., July 31, 1908, p. 115.)

With highly superheated steam involving temperatures of 600 degrees F. or more the poppet-valve type of engine (Figs. 220, 221) is ordinarily employed, though balanced piston valves are not uncommon. The poppet valve is not distorted by heat and requires no lubrication. In Europe these engines have been brought to a high state of efficiency, but have not been generally adopted in this country.

119. Properties of Superheated Steam. — The laws governing superheated vapors, like those governing saturated vapors, are not rational and deducible from a few fundamental experiments, but are more or less empirical in character. However, although the numerical values of the various quantities are based on the results of experiments, they permit of accurate mathematical formulation. Thus the following equations, derived by Prof. Goodenough ("Principles of Thermodynamics," 1911) and based upon the experiments of Knoblauch, Jakob, and Linde, give results which agree substantially with standard superheated steam tables.

T = absolute temperature of the superheated steam, degrees F.

p = absolute steam pressure, pounds per square inch.

λ = total heat, B.t.u. per pound.

u = intrinsic energy, B.t.u. per pound.

n = entropy.

C_p = true specific heat.

v = specific volume, cubic feet per pound.

$$\lambda = T\,(0.367 + 0.00005\,T) - p\,(1 + 0.0003\,p)\,\frac{C}{T^5} - 0.0163p + 886.7, \tag{39}$$

in which $\log C = 13.72511$.

$$u = T\,(0.2566 + 0.00005\,T) - \frac{Cp}{T^5}\,(1 + 0.00024\,p) + 886.7, \tag{40}$$

in which $\log C = 13.64593$.

$$n = 0.8451 \log T + 0.0001\, T - 0.2542 \log p - p\,(1 + 0.003\, p)\,\frac{C}{T^6} - 0.3964, \tag{41}$$

in which $\log C = 13.64593$.

$$C_p = 0.367 + 0.0001\, T + p\,(1 + 0.0003\, p)\,\frac{C}{T^6}, \tag{42}$$

in which $\log C = 14.42408$.

$$v = 0.5963\,\frac{T}{p} - (1 + 0.0006\, p)\,\frac{47{,}795 \times 10^9}{T^5} - 0.088, \tag{43}$$

in which $\log C = 13.64593$.

Wm. J. Goudie (Engng., July 1, 1910) gives the following simple equation for determining the specific volume which gives results sufficiently accurate for many engineering purposes.

$$v = v_1\,(1 + 0.0016\, t), \tag{44}$$

in which

v_1 = specific volume of saturated steam,

t = degree of superheat, degrees F.

The *mean* specific heat may be obtained by subtracting the total heat of the saturated steam from that of the superheated steam and dividing the difference by the degree of superheat.

Practically all commercial engineering problems are most conveniently solved by means of superheated steam tables and diagrams, and recourse to formulas is seldom necessary.

The curves shown in Fig. 115 give the *true* specific heats and those in Fig. 116 the *mean* specific heats of superheated steam for all pressure and temperature ranges likely to occur in practice. These curves are taken from Goodenough's "Principles of Thermodynamics," 1911, and are probably more accurate than those found in Marks and Davis' Steam Tables, though the difference is small.

The Mollier diagram for superheated and saturated steam is reproduced and described in Appendix L.

A Discussion on Certain Thermal Properties of Steam: Prof. G. A. Goodenough, Jour. A.S.M.E., April, 1912.

The Battle of the Superheats: R. H. Smith, Engineer, Dec. 22, 1911.

Duchesne's Experiments on Superheat: V. Dwelshauver's-Dery, Power, July 23, 1912, p. 110.

Complete Discussion of the Work of Various Investigators: Dr. H. N. Davis, Proc. Am. Academy of Arts and Sciences, Vol. 45, p. 267, 1910.

120. Superheaters. — Superheaters are manufactured by practically all boiler builders, the characteristics of the boiler being embodied to a large extent in the design of the superheater. The superheater may

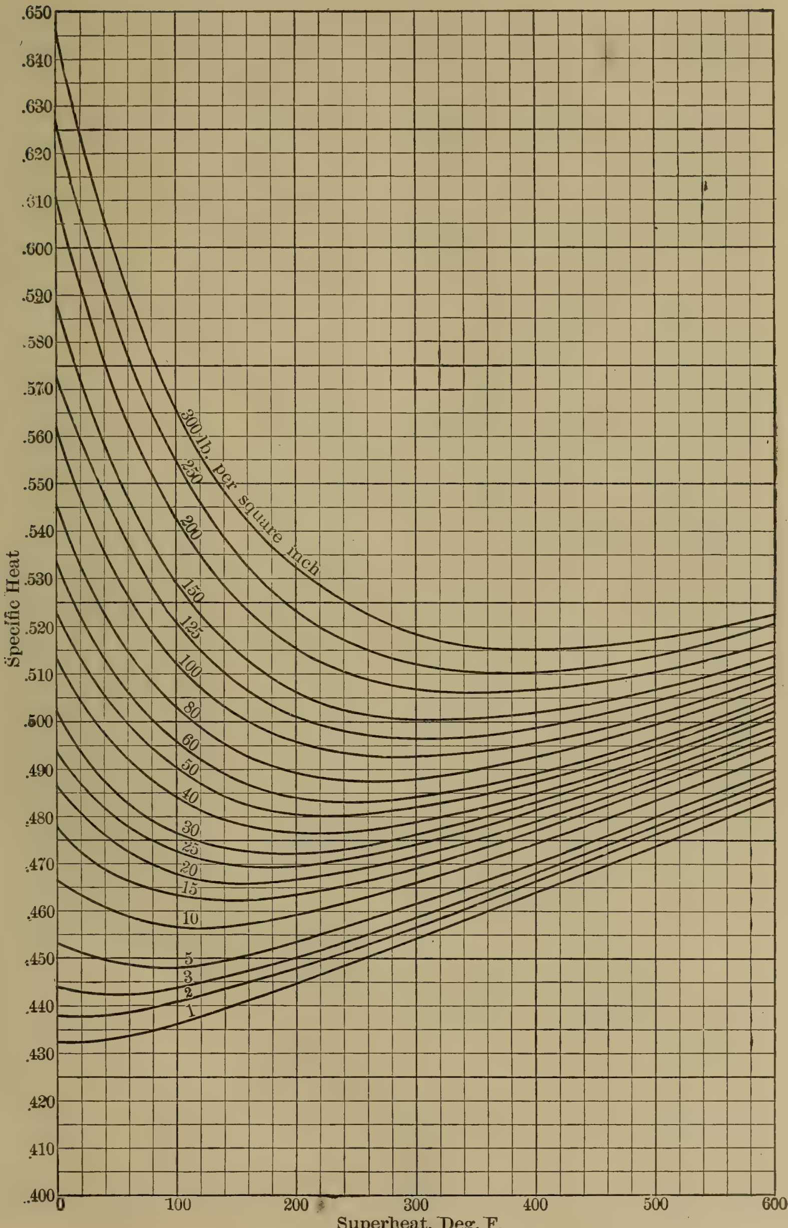


Fig. 115. True Specific Heat of Superheated Steam.

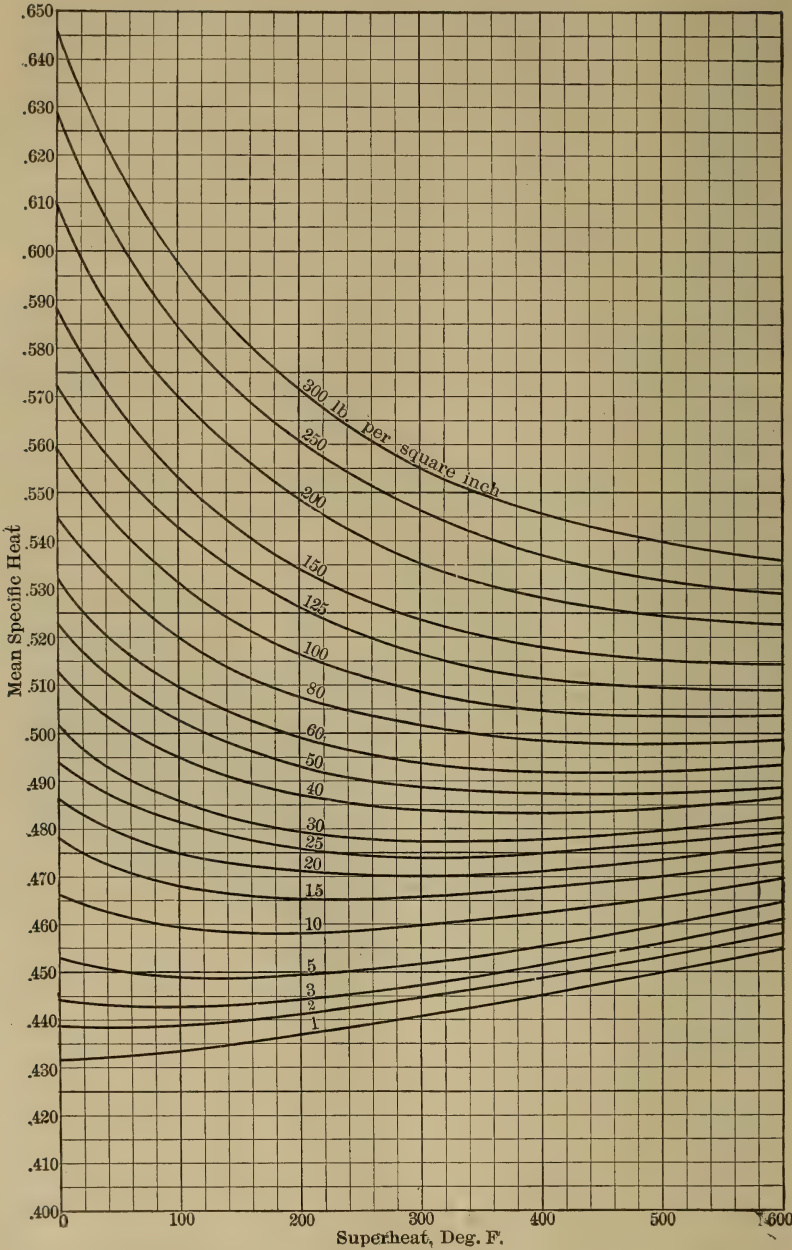


FIG. 116. Mean Specific Heat of Superheated Steam.

TABLE 34.

MEAN SPECIFIC HEAT OF SUPERHEATED STEAM.

(Computed from Marks and Davis' Steam Tables.)

Absolute Pressure, Pounds per Square Inch.	Degrees of Superheat, Fahr. 10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250	300
1	.452	.452	.453	.453	.454	.454	.454	.455	.455	.455	.455	.455	.455	.456	.456	.456	.456	.456	.456	.457
5	.460	.460	.460	.460	.460	.460	.459	.459	.459	.459	.459	.459	.459	.459	.459	.459	.460	.460	.460	.460
10	.465	.465	.465	.465	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.465	.465	.465	.465
15	.470	.470	.470	.470	.470	.470	.469	.469	.469	.469	.469	.469	.469	.469	.469	.468	.468	.468	.468	.468
20	.475	.475	.475	.475	.474	.474	.474	.474	.473	.473	.473	.473	.473	.472	.472	.472	.472	.472	.471	.471
25	.480	.480	.480	.479	.479	.479	.478	.478	.478	.477	.477	.477	.477	.477	.476	.476	.476	.475	.475	.475
30	.485	.485	.484	.484	.484	.483	.483	.483	.483	.482	.482	.481	.481	.481	.480	.480	.479	.479	.478	.477
35	.490	.490	.489	.489	.488	.488	.487	.487	.486	.486	.486	.485	.485	.484	.484	.483	.482	.482	.481	.480
40	.495	.494	.494	.493	.492	.492	.491	.491	.490	.490	.489	.489	.488	.488	.487	.486	.485	.484	.483	.482
50	.509	.508	.507	.506	.504	.503	.501	.500	.499	.498	.497	.496	.496	.495	.494	.493	.491	.490	.489	.487
75	.540	.535	.533	.530	.528	.525	.523	.521	.519	.517	.515	.513	.512	.511	.509	.506	.504	.502	.500	.497
100	.570	.560	.556	.552	.550	.546	.542	.540	.537	.534	.531	.528	.526	.524	.522	.518	.515	.512	.509	.505
125	.590	.585	.580	.575	.570	.565	.560	.556	.552	.548	.545	.542	.539	.537	.534	.532	.524	.520	.517	.510
150	.620	.615	.606	.597	.592	.585	.579	.572	.566	.562	.558	.554	.550	.547	.544	.539	.533	.528	.524	.518
175	.660	.645	.633	.622	.614	.605	.597	.589	.582	.577	.572	.566	.562	.558	.555	.548	.541	.535	.531	.524
200	.690	.675	.657	.645	.634	.623	.614	.605	.596	.590	.584	.578	.572	.568	.563	.556	.548	.542	.537	.529
225	.730	.710	.686	.672	.656	.643	.631	.620	.611	.603	.595	.589	.583	.578	.573	.562	.553	.547	.543	.535
250	.770	.740	.712	.695	.678	.663	.648	.635	.624	.615	.606	.599	.593	.587	.582	.570	.562	.555	.549	.540
275	.800	.770	.746	.722	.700	.681	.664	.650	.638	.629	.620	.610	.602	.596	.590	.578	.569	.561	.555	.545
300	.850	.805	.773	.740	.724	.702	.683	.666	.652	.641	.630	.621	.612	.605	.599	.586	.576	.568	.561	.550

TABLE 35.

SPECIFIC VOLUME OF SUPERHEATED STEAM.

Cubic Feet per Pound.

(Knoblauch, Linde and Klebe.)

Absolute Pressure, Pounds per Square Inch.	Degrees of Superheat, Fahr.																			
	0	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250
1	333.0	339.0	345.0	352.0	358.0	364.0	370.0	376.0	331.0	388.0	394.0	400.0	406.0	412.0	418.0	424.0	439.0	454.0	468.0	483.0
5	73.3	74.6	75.9	77.2	78.4	79.7	80.9	82.1	83.3	84.5	85.7	86.9	88.2	89.4	90.6	91.8	94.8	97.8	101.0	104.0
10	38.4	39.0	39.7	40.3	40.9	41.5	42.2	42.8	43.4	44.0	44.6	45.2	45.8	46.4	47.0	47.7	49.2	50.7	52.2	53.7
15	26.3	26.7	27.1	27.6	28.0	28.4	28.8	29.2	29.6	30.0	30.5	30.9	31.3	31.7	32.1	32.5	33.5	34.5	35.5	36.6
20	20.1	20.4	20.7	21.0	21.4	21.7	22.0	22.3	22.6	22.9	23.3	23.6	23.9	24.2	24.5	24.8	25.5	26.3	27.0	27.8
25	16.3	16.6	16.8	17.1	17.3	17.6	17.8	18.1	18.4	18.6	18.8	19.1	19.4	19.6	19.8	20.1	20.7	21.3	21.9	22.6
30	13.7	14.0	14.2	14.4	14.6	14.8	15.0	15.2	15.5	15.7	15.9	16.1	16.3	16.5	16.7	16.9	17.4	18.0	18.5	19.0
35	11.9	12.1	12.3	12.5	12.7	12.9	13.0	13.2	13.4	13.6	13.7	13.9	14.1	14.3	14.5	14.6	15.0	15.5	15.9	16.4
40	10.5	11.2	11.4	11.5	11.7	11.9	12.1	12.2	12.4	12.6	12.7	12.9	13.1	13.2	13.4	13.6	14.0	14.4	14.8	15.2
50	8.5	8.7	8.8	8.9	9.0	9.2	9.3	9.4	9.6	9.7	9.8	9.9	10.1	10.2	10.3	10.5	10.8	11.1	11.4	11.7
75	5.81	5.91	6.00	6.10	6.19	6.28	6.37	6.46	6.55	6.64	6.73	6.82	6.91	7.00	7.09	7.17	7.38	7.60	7.81	8.02
100	4.43	4.51	4.58	4.65	4.72	4.79	4.86	4.93	5.00	5.07	5.14	5.21	5.27	5.34	5.41	5.47	5.63	5.80	5.96	6.12
125	3.58	3.64	3.70	3.76	3.82	3.88	3.94	4.00	4.06	4.11	4.17	4.22	4.28	4.33	4.39	4.45	4.58	4.71	4.84	4.97
150	3.01	3.06	3.11	3.17	3.22	3.27	3.32	3.37	3.42	3.46	3.51	3.56	3.61	3.66	3.70	3.75	3.86	3.97	4.08	4.19
175	2.60	2.65	2.69	2.74	2.78	2.83	2.87	2.91	2.96	3.00	3.04	3.08	3.12	3.16	3.20	3.24	3.34	3.44	3.53	3.63
200	2.29	2.33	2.37	2.41	2.45	2.49	2.53	2.57	2.61	2.64	2.68	2.72	2.76	2.79	2.83	2.86	2.95	3.04	3.12	3.21
225	2.05	2.09	2.12	2.15	2.19	2.23	2.26	2.30	2.33	2.37	2.40	2.43	2.47	2.50	2.53	2.57	2.64	2.72	2.80	2.88
250	1.85	1.88	1.91	1.95	1.98	2.02	2.05	2.08	2.11	2.14	2.19	2.21	2.24	2.27	2.30	2.33	2.40	2.47	2.54	2.61
275	1.69	1.72	1.75	1.78	1.81	1.84	1.87	1.90	1.93	1.96	1.99	2.02	2.04	2.07	2.10	2.13	2.19	2.26	2.32	2.39
300	1.55	1.58	1.60	1.63	1.66	1.69	1.72	1.75	1.78	1.80	1.83	1.86	1.88	1.91	1.94	1.96	2.02	2.09	2.15	2.21

be independently fired or placed in the boiler setting. In the latter arrangement the superheater may be located between the furnace and the heating surface, as in Fig. 49, at the end of the heating surface, as in Fig. 125, or at some intermediate point, as in Figs. 119 and 121. Since the absorption of heat depends chiefly upon the average temperature difference between the gases and the steam and the extent of superheating surface, the required degree of superheat may be obtained from a small extent of heating surface in the furnace, a large amount in the rear of the heating surface or a proportionate amount in intermediate locations. In a general sense the sum of the boiler heating surface and superheating surface per boiler horse power is practically the same for any degree of superheat. The cost of a superheated steam boiler is approximately equal to that of a saturated steam boiler since the superheated plant has less steam to generate. The requirements of a successful superheater are:

1. Security of operation, or minimum danger of overheating.
2. Economical use of heat applied.
3. Provision for free expansion.
4. Disposition so that it may be cut out without interfering with the operation of the plant.
5. Provision for keeping tubes free from soot and scale.

Superheaters may be *separately fired* or *indirectly fired.* The advantages of the separately fired superheater are:

1. The degree of superheat may be varied independently of the performance of the boiler.
2. It may be placed at any desired point.
3. Repairs are readily made without shutting down the boiler.

Some of the disadvantages are:

1. It requires separate attention.
2. Saturated steam only can be furnished to the prime movers in case of a breakdown to the superheater.
3. Extra piping is required.
4. Extra space is required.

The indirectly fired superheater arranged in the boiler setting has the advantage of:

1. Lower first cost.
2. Higher operating efficiency.
3. Minimum attention.
4. Minimum space requirements.

As ordinarily installed the indirectly fired superheater is subject to the fluctuating temperatures of the furnace so that forcing the boiler has a similar effect on the superheater. In some cases the superheater adjusts itself automatically to the load requirements maintaining a constant degree of superheat at all loads, but in most cases the degree of superheat increases with the load, see Fig. 131.

Standard central station practice in this country favors the superheater contained within the boiler setting.

121. Babcock and Wilcox Superheater. — Figs. 117 and 118 show the application of superheating coils to a Babcock and Wilcox boiler illustrating the usual location of the indirectly fired type. The coils are made

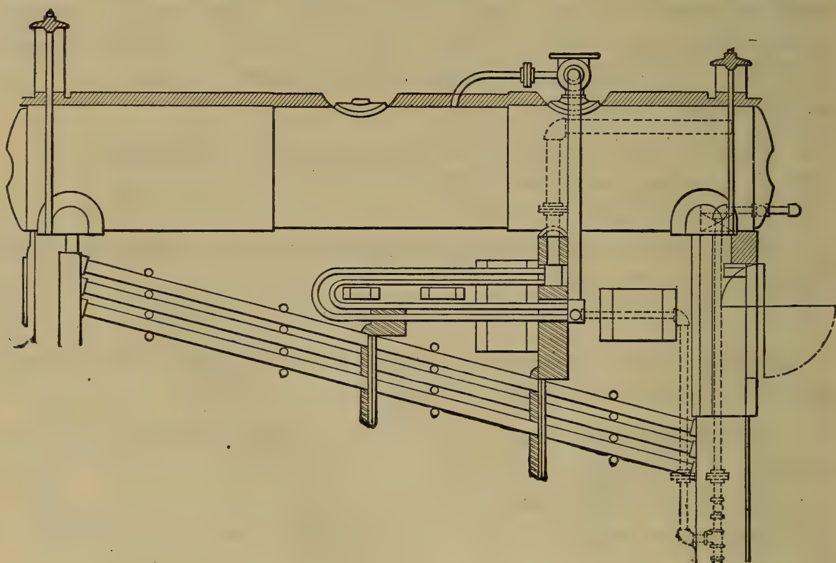


FIG. 117. Babcock and Wilcox Superheater.

of 2-inch No. 8 gauge seamless steel tubes expanded into forged steel headers, the upper one receiving the saturated steam from the boiler and the lower one the superheated steam after it has traversed the superheater tubes. A small pipe connects the lower manifold with the water space of the boiler by means of which the superheater may be cut out if desired, or flooded when starting up. Any steam formed in the superheater tubes is returned into the boiler drum through the collecting pipe, which, when the superheater is at work, conveys saturated steam into the upper manifold. When steam pressure has been attained the superheater is thrown into action by draining the water away from the manifolds and opening the superheater stop valve. The tubes are free at one end and the manifolds are not rigidly con-

nected with each other, thus avoiding expansion strains. With the proportion of superheating surface to boiler surface ordinarily adopted the steam is superheated from 100 to 150 degrees F.

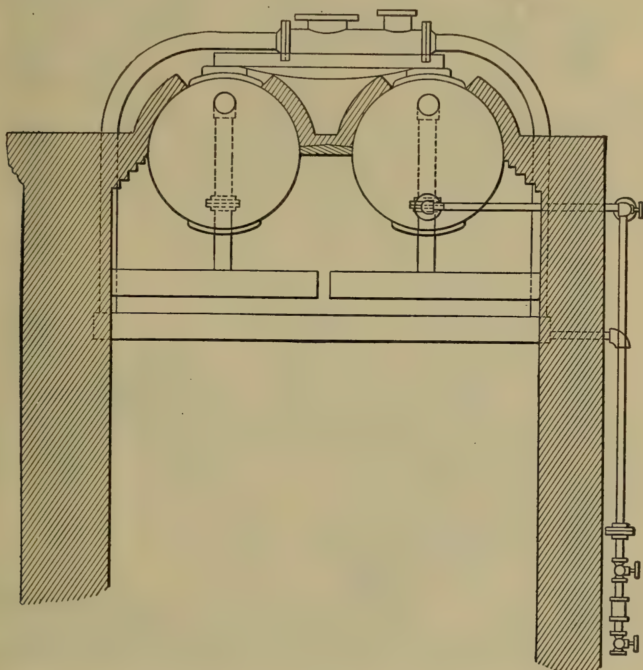


FIG. 118. Babcock and Wilcox Superheater.

122. Stirling Superheater. — This superheater consists of two drums, Fig. 120, connected by seamless drawn tubes two inches in diameter. It may take the place of the middle bank of tubes in the Stirling boiler, as shown in Fig. 119, or be installed in the final pass of the gases in the back of the boiler. The drums around the tubes are protected from intense heat by asbestos cement. A pipe connecting the front drum of the boiler with the lower drum of the superheater permits the coils to be flooded in starting up or when the superheater is not needed. In this case the superheater acts as additional boiler-heating surface. The upper drum is divided into three and the lower into two compartments. The tubes are arranged with alternately wide and narrow spacing, so that any tube may be removed without disturbing the rest. The flow of steam is indicated by arrows.

123. Foster Superheater. — Fig. 121 shows the application of a Foster superheater to a Babcock and Wilcox boiler. This device consists of cast-iron headers joined by a bank of straight parallel seamless

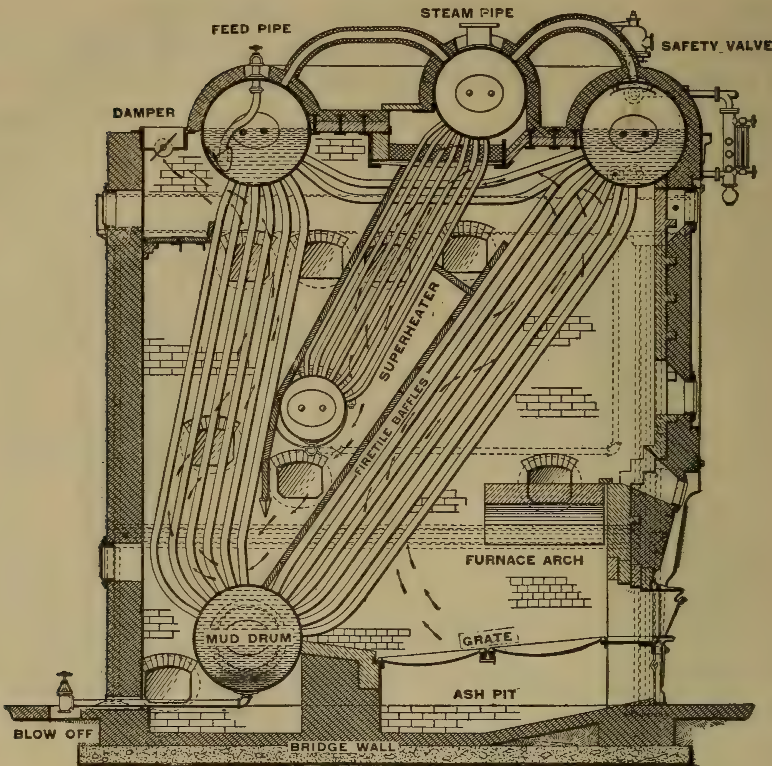


Fig. 119. Stirling Superheater.

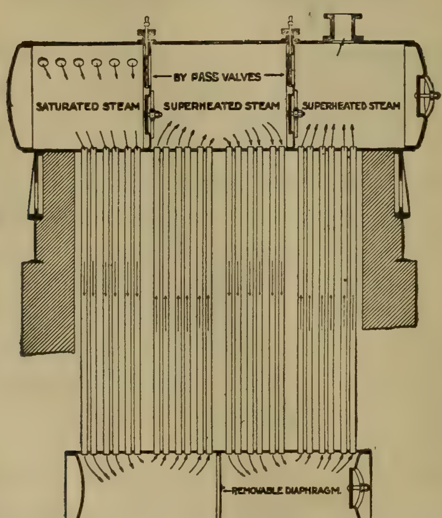


Fig. 120. Arrangement of Tubes; Stirling Superheater.

drawn-steel tubes, each tube being encased in a series of annular flanges placed close to each other and forming an external cast-iron covering of large surface. The protection afforded by this external covering is ample to prevent damage from overheating during the process of steam

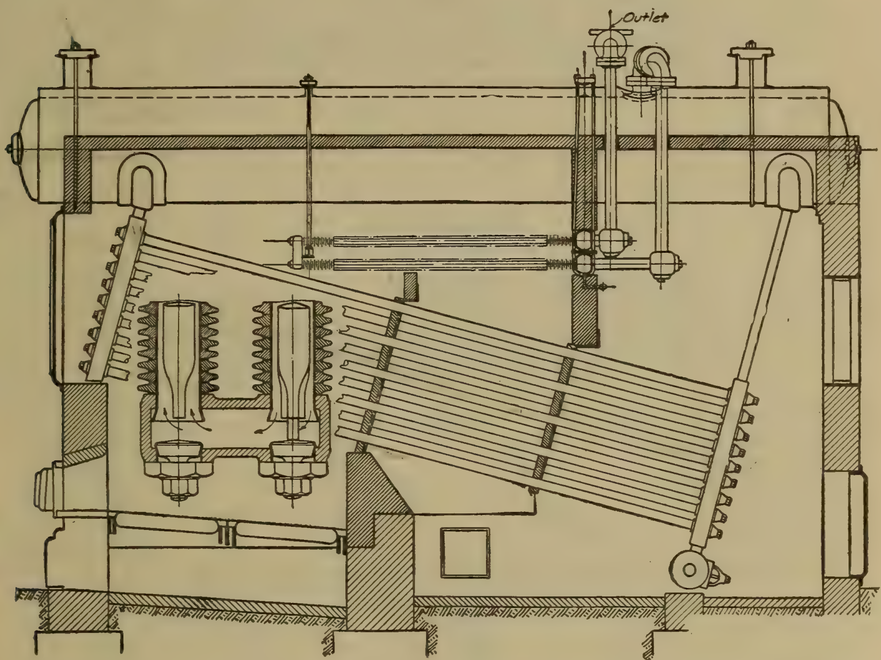


FIG. 121. Foster Superheater in Babcock and Wilcox Boiler.

raising, and flooding devices are unnecessary. The tubes are double, the inner tube serving to form a thin annular space through which the steam passes as indicated. Caps are provided at the end of each element for inspection and cleaning purposes. Foster superheaters are more costly than plain-tube superheaters, but are longer lived and offer a much larger heating surface in proportion to the space occupied.

Fig. 124 shows a Foster superheater arranged for independent firing.

The "Schwoerer" superheater, which is somewhat similar in external appearance to the Foster, differs from it considerably in detail, the heating surface being made up of suitable lengths of cast-iron pipe ribbed outside circumferentially and inside longitudinally. The ends of the pipes are flanged and connected by cast-iron U-bends. The intention is to provide ample heating surface internally and externally, with a compact apparatus.

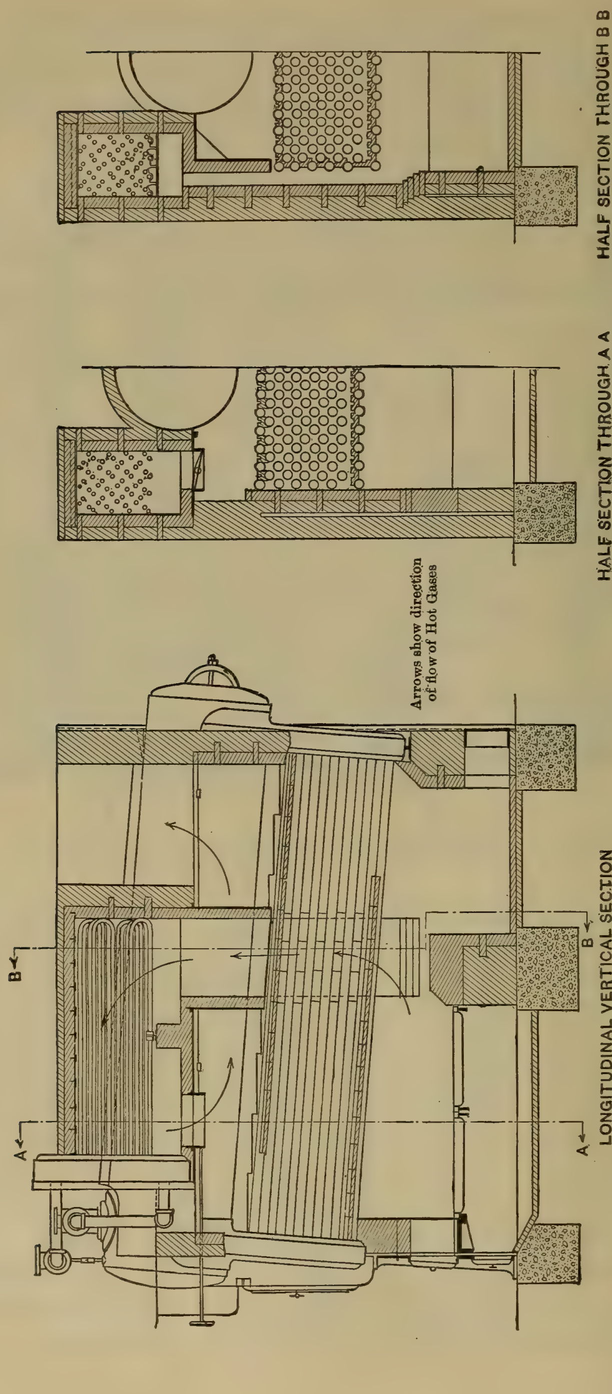


Fig. 122. Heine Superheater.

124. Heine Superheater. — Fig. 122 shows the application of a Heine superheater to a Heine boiler, illustrating the installation of a superheater within the boiler setting but entirely separated from the main gas passages. The superheater consists essentially of a number of $1\frac{1}{2}$-inch seamless steel tubes, bent to U-shape and expanded into a header box of the same type of construction as the standard Heine boiler water leg. The interior of this box is divided into three compartments by light sheet-iron diaphragms, so as to deflect the current of steam through the tubes. The superheater chamber is located above the steam drum as indicated. The gases of combustion are led to the superheater chamber through a small flue built in the side walls of the setting. A damper placed at the outlet of the flue controls the flow of gases and regulates the degree of superheat. No provision is necessary for flooding the superheating coils since the gases may be entirely diverted from the heating surface. Soot accumulations are readily removed by introducing a soot blower through the hollow stay bolts.

125. Independently Fired Superheaters. — The Schmidt superheater, Fig. 123, consists of two nests of coils, A and D, of equal size and dimensions, connected to cast-iron headers O and I. Saturated steam enters the first nest of coils through C and passes into header O. From O the steam, which is now dried, and partly superheated, flows through the cast-iron pipe E to header I, and thence through the second nest of coils into header adjoining O, and through pipe R to the engine. In chamber D the steam and gases flow on the counter-current and in chamber A on the concurrent principle. This combination permits of a low flue temperature and high steam temperature without subjecting the tubes to an excess of heat as would be the case if the steam left the coils A at header I, where the furnace gases are the hottest. A steam temperature of 750 degrees F. and a flue temperature of 450 degrees F. are easily maintained with this apparatus. A mercury pyrometer T is fitted where the superheated steam enters the discharge pipe R. A thermometer cup L permits of checking the pyrometer by means of a nitrogen-filled thermometer. Each coil can be taken out separately and a new one put in without removing the others or dismantling the plant. Water produced by condensation while the superheater is inoperative collects in the bottom header N and escapes through a drain cock. If the steam supply should be suddenly shut off, the air door P is opened automatically by weight K. As soon as steam begins to flow it raises the weight through the opening of valve C and the door closes. The Schmidt superheater when arranged in the flue has practically the same construction as the independently fired.

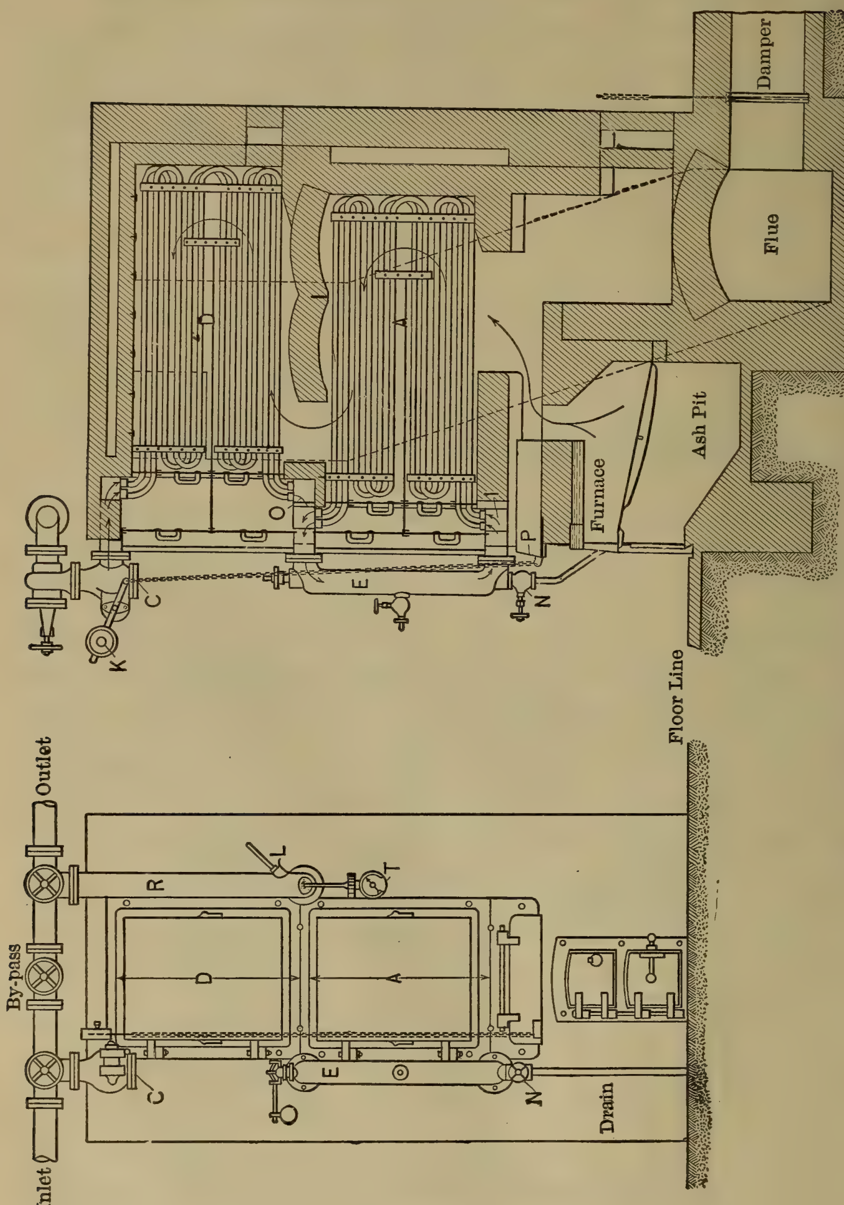


FIG. 123. Schmidt Independently Fired Superheater.

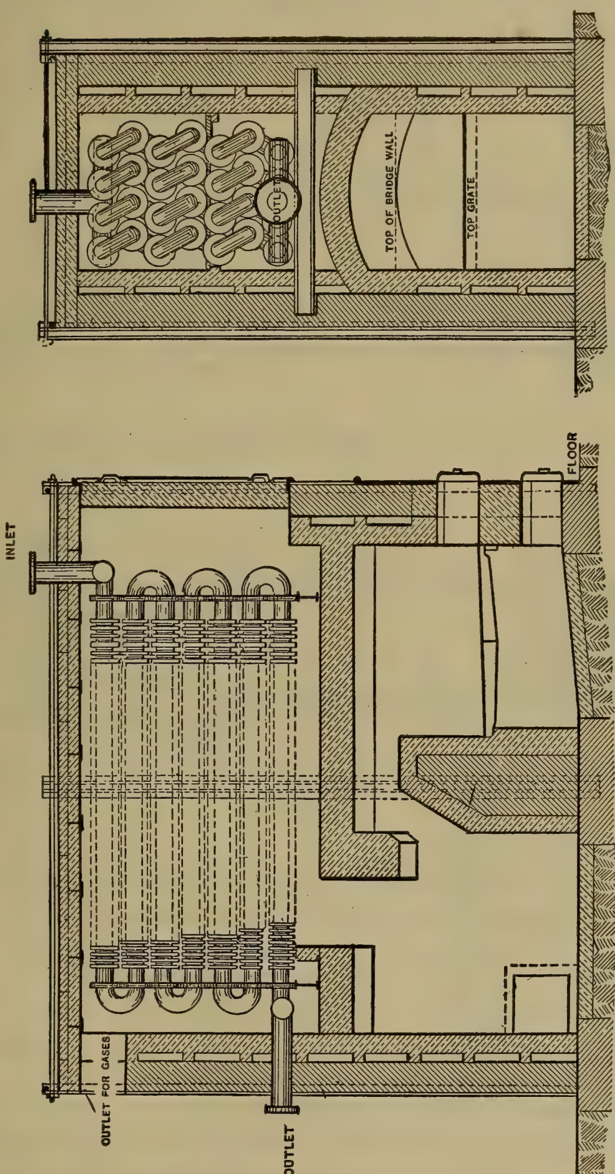


FIG. 124. Foster Independently Fired Superheater.

Fig. 125 shows a combination of Schmidt superheater, economizer, and feed-water heater which finds much favor with engineers on the continent.

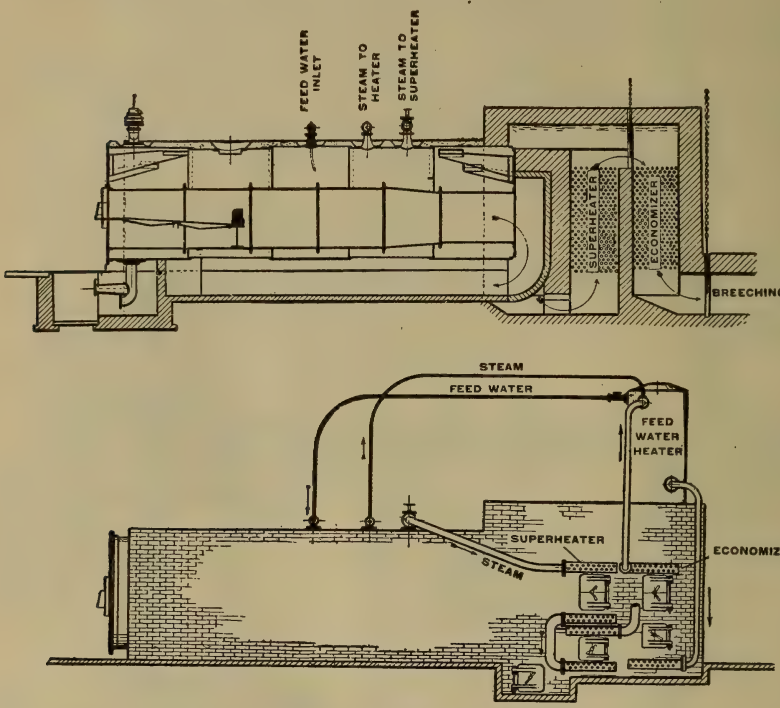


Fig. 125. Schmidt System of Combined Superheater, Feed-water Heater, and Economizer.

126. Luckenbach Superheater. — Fig. 126 shows a section through a Luckenbach superheater illustrating an extremely simple and effective device for superheating small quantities of steam up to very high temperatures. It consists essentially of a single coil of extra heavy $1\frac{1}{2}$-inch steel pipe imbedded within the walls of a cylindrical casting. The coil is not welded to the casting but is free to expand and contract independently. The furnace illustrated in Fig. 126 is designed for hard coal or coke. The apparatus is compact and durable and no provision is necessary for flooding the coils. The steam supply may be cut off entirely with a furnace full of incandescent fuel without burning out the coils. The following results were obtained from capacity tests of a Luckenbach superheater 30 inches in diameter by 20 inches in height, as installed in the Mechanical Engineering Laboratory of the Armour Institute of Technology.

CAPACITY TEST OF A 30-INCH LUCKENBACH SUPERHEATER.

Lineal feet of superheater coil........................ 28 feet.
Internal heating surface of furnace walls.............. 7.5 square feet.
Grate area.................................... 1.5 square feet.

Steam Pressures, Lbs. per Sq. In. Gauge.		Moisture in Steam Entering Superheater, Per Cent.	Steam Temperatures, Degrees F.			Weight of Steam Flowing, Lbs. per Hr.	Weight of Coke Fired, Lbs. per Hr
Entering Super-heater.	Leaving Super-heater.		Entering Super-heater.	Leaving Super-heater.	Degrees of Super-heat.		
50	40	0.9	298	587	300	739.6	17.7
50	45	0.9	298	618	325	678.5	17.1
50	46	1.0	298	648	354	585.4	17.1
50	46.5	1.1	298	700	406	505.5	17.6
50	47	2.3	298	760	465	371.0	17.6
50	47.5	1.4	298	790	495	341.0	17.7
70	67	0.7	316	803	490	359.0	17.7
70	66	1.6	316	705	392	484.0	*10.2

Damper wide open throughout all tests, no attempt being made to obtain high furnace efficiency.
* Economy test, damper throttled.

127. Materials used in Construction of Superheaters. — Most superheaters are constructed either of wrought iron, mild steel, cast iron, or cast steel, the latter material having the advantage of not being damaged by any temperature to which it is likely to be subjected, which does away with the necessity of damper mechanisms and simplifies the installation. On the other hand, cast-metal superheaters are usually ribbed after the fashion of an air-cooled gas engine, and are, therefore, very heavy and thick walled, necessitating a higher temperature for the same useful effect than in the case of the wrought-iron construction, but have the advantage of minimizing fluctuation of steam temperature which would otherwise be caused by a wide variation in temperature of furnace. One of the most successful cast-metal heaters is of European design and is constructed of a special alloy known as "Schwoerer" iron. Table 36 gives the yearly cost of repairs to piping and necessary brickwork for a number of installations equipped with cast-metal superheaters of the "Schwoerer" type.

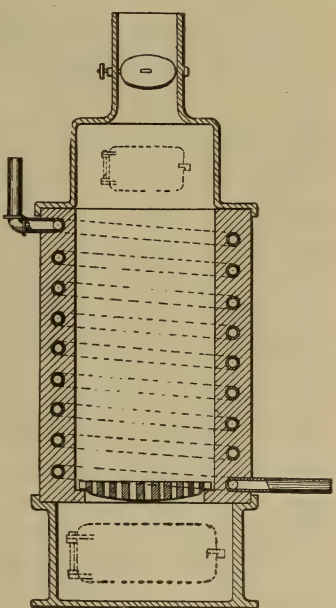


FIG. 126.
Section through Luckenbach Independently Fired Superheater.

Wrought iron and mild steel offer the advantage of lightness, ease of construction, and low first cost, but cannot be exposed to very high temperatures without injury, and consequently provision must be made for diverting the direction of the heated gases or for flooding the coils while the boiler is being warmed before steam is generated.

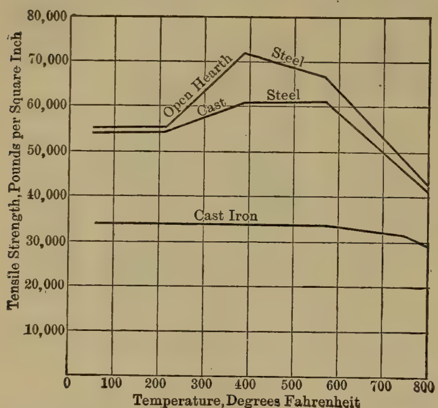


Fig. 127. Effect of Temperature on Strength of Materials.

The effect of temperature on superheater materials is shown in Fig. 127. It will be seen that the tensile strength drops off very rapidly for temperatures beyond 650 degrees F. Because of this rapid decrease in tensile strength of materials with the increase in temperature, steam is seldom superheated to temperatures above 850 degrees F.

For further information pertaining to the effect of temperature on various metals, consult "The Effect of High Temperatures on the Physical Properties of Some Metals and Alloys"; The Valve World, Jan., 1913, published by the Crane Co., Chicago.

Ordinary cast-iron valves and fittings have shown permanent increase in dimensions under high superheat and in numerous instances have failed altogether, but sufficient data are not available to prove conclusively the unreliability of cast iron if the iron mixture is properly compounded and the necessary provision is made for expansion and contraction. Authorities are of the opinion that the failure of cast-iron fittings is due more to fluctuations in temperature than to the actual high temperature itself and cite numerous cases where ordinary cast-iron fittings under uniform temperature conditions are giving satisfaction with highly superheated steam. Notwithstanding the claims that cast iron properly compounded is a perfectly reliable metal for fittings, engineers are inclined to use cast or forged steel, at least in this country. See "Effect of Superheated Steam on Cast Iron and Steel," Trans. A.S.M.E., Vol. 31, 1909, p. 989.

128. Extent of Superheating Surface. — The required extent of superheating surface for any proposed installation depends upon: (1) the degree of superheat to be maintained; (2) the velocity of the steam and gases through the superheater; (3) the character of the superheater; (4) the weight of steam to be superheated; (5) the moisture in the wet steam; (6) the temperature of the gases entering and leav-

TABLE 36.

AVERAGE YEARLY EXPENSES FOR REPAIRS FOR CAST-IRON SUPERHEATERS (" SCHWOERER " TYPE) AS OBTAINED FROM SIX PLACES OF INSTALLATION.

(Otto Berner, *Power*, August, 1904.)

Number.	Length of Time of Installation. Years.	Average Daily Use. Hours.	Place of Installation of Superheater with Reference to Boiler.	Average Temp. of Gases immediately in Front of the Superheater Surface. Degrees F.	Average Steam Pressure. Pounds per Square Inch (Absolute).	Average Temp. of Steam. Degrees F.	No. of Elements for One Superheater.	Length of One Element in Feet.	Average Yearly Cost of Repairs.	
									1 Superheater.	For 1 Foot of Element.
1	8	11	Directly heated	....	99.56	518°	12	10.5	$71.40	$0.61
2	5	12	Directly behind the fire bridge	....	106.672	....	4	9.843	$14.09	$0.37
3	10	11	*a.* Directly behind the fire bridge .... *b.* Behind the first flue.	.... 716°–842°	78.232 113.79	.... 442.4°	2 6	8.202 9.843	11.90 0	$0.68 0
4	11	12	Behind the first flue	....	85.34	392°–572°	6	9.843	$8.33	$0.15
5	6	24	Behind the first flue (fire-tube boiler)	....	99.56 & 156.46	482°	8 to 10	6.562	$5.95	$0.08
6	7	11	Directly behind the fire bridge	....	170.68	500°–572°	....	....	0	0
Average Results	7.7	13.5	....	....	113.79	493°	6.9	5.25	$15.95	$0.28

ing the superheater; (7) the conductivity of the material, and (8) cleanliness of the tubes.

Since the heat absorbed by the steam in the superheater is equal to that given up by the products of combustion, neglecting radiation, this relationship may be expressed

$$SUd = Wc\,(t_1 - t_2), \tag{45}$$

in which

S = square feet of superheating surface per boiler horse power,

U = coefficient of heat transmission, B.t.u. per hour per degree difference in temperature,

d = mean temperature difference between the steam and heated gases, degrees F.,

W = weight of gases passing through the superheater per boiler-horse-power hour,

c = mean specific heat of the gases,

t_1 = temperature of the gases entering superheater, degrees F.,

t_2 = temperature of the gases leaving superheater, degrees F.

Transposing equation (45),

$$S = \frac{Wc\,(t_1 - t_2)}{Ud}. \tag{46}$$

The heat transfer from the products of combustion to the steam may also be expressed

$$SUd = wc'\,(t_s - t), \tag{47}$$

in which

w = weight of steam passing through the superheater, pounds per boiler-horse-power hour,

c' = mean specific heat of the superheated steam,

t_s = temperature of the superheated steam, degrees F.,

t = temperature of the saturated steam, degrees F.,

S, U, and d as in equation (45).

For wrought-iron or mild steel tubes U varies as follows:

U = 1 to 3 for superheaters located at the end of the heating surface,

= 3 to 5 for superheaters located between the first and second pass of water tube boilers,

= 8 to 12 for superheaters located immediately above the furnace in stationary boilers, in the smoke box of locomotive boilers, and in separately fired furnaces.

General practice allows $\frac{1}{4}$ to $\frac{1}{5}$ square foot of superheating surface per boiler horse power for mild steel, superheater located in the furnace; from 2 to 2.5 square feet of surface at the end of the first pass, and from 3 to 4 square feet at the end of the heating surface for superheats of from 100 to 150 degrees F., boiler pressure 150 pounds absolute.

The Foster Superheater Company allows 6 B.t.u. per lineal foot per degree difference in temperature for their "two-inch" element where the average temperature of the gases is about twice the mean temperature of the steam.

For all engineering purposes d may be determined with sufficient accuracy from the relationship

$$d = \frac{t_1 + t_2}{2} - \frac{t_s + t}{2}.$$

Notations as in equations (45) and (47).

An empirical formula for determining the extent of superheating surface in connection with indirect superheaters which appears to conform with practice is derived by substituting

$$U = 3, \quad d = t' - \frac{t_s + t}{2}, \quad w = 30, \quad c' = 0.5,$$

in equation (47) [J. E. Bell, Trans. A.S.M.E., 29–267]. Thus:

$$S \times 3\left(t' - \frac{t_s + t}{2}\right) = 30 \times 0.5 \times (t_s - t),$$

from which

$$S = \frac{10\,(t_s - t)}{2\,t' - t_s - t}; \tag{48}$$

t' (the mean temperature of the product of combustion where the superheater is located) may be approximated from equation

$$\frac{1}{(t' - t)^{0.16}} = 0.172\,H + 0.294, \tag{49}$$

in which

H = the per cent of boiler-heating surface between the point at which the temperature is t and the furnace,

t as in (48).

Equation (49) is based upon the assumption that the heat transferred from the gases to the water is directly proportional to the difference in temperature; that the furnace temperature is 2500 degrees F.; flue temperature 500 degrees F.; steam pressure 175 pounds per square inch gauge; one boiler horse power is equivalent to 10 square feet of water-heating surface.

Example: What extent of heating surface is necessary to superheat saturated steam at 175 pounds gauge pressure, 200 degrees F., if the superheater is placed in the boiler setting where the gases have already traversed 40 per cent of the water-heating surface?

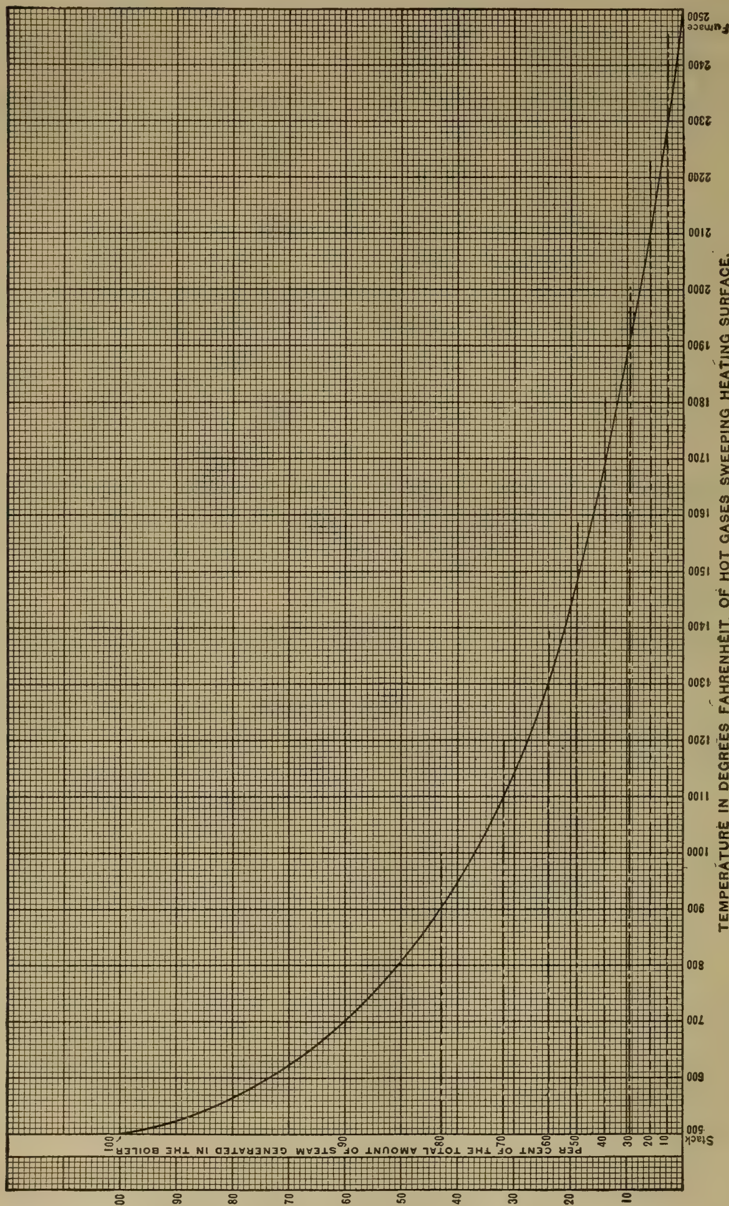


FIG. 128. Curve showing Relation between Gas Temperature and Extent of Heating Surface passed over.

Substitute $H = 0.4$ and $t = 378$ in equation (49),

$$\frac{1}{(t' - 378)^{0.16}} = 0.172 \times 0.4 + 0.294,$$

from which $$t' = 950.$$

Substitute $t' = 950$, $t_s = 578$, and $t = 378$ in equation (48),

$$S = \frac{10\,(578 - 378)}{2 \times 950 - 578 - 378}$$
$$= 2.12 \text{ square feet.}$$

The curve in Fig. 128 was plotted from equation (49) and gives a ready means of determining t' and of observing the law governing heat absorption by the boiler between furnace and breeching. The abscissas represent the temperatures of the hot gases at different points in their path between furnace and breeching. The ordinates represent (1) the per cent of boiler-heating surface passed over by the hot gases, and (2) the per cent of the total heat generated which is absorbed by this heating surface.

In the use of equation (49) the probability of error is greatest when considering a point near the furnace, since large quantities of heat are transmitted to the tubes by radiation from the fuel bed which are not taken account of. For most practical purposes the assumption is sufficiently accurate.

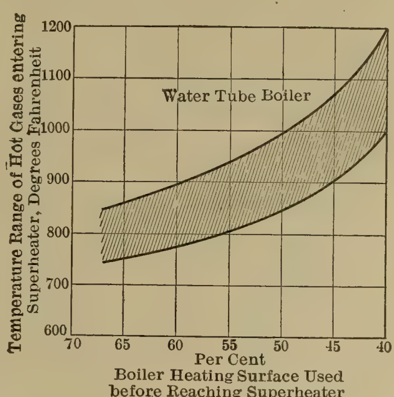


FIG. 129. Temperature Range of Gases in Superheater.

Fig. 129 gives the probable temperature range of gases entering superheater after passing over a given per cent of boiler-heating surface and Fig. 130 shows the relation between superheating surfaces and boiler heating surface. (See Power, Nov. 7, 1911, p. 696.)

It will be found that the boiler-heating surface per boiler horse power will be decreased in almost the same proportion that the superheating surface is increased, so that the sum of the boiler-heating surface and superheating surface per boiler horse power will be very nearly the same for any degree of superheat.

For the application of the curve in Fig. 128 to the design of direct and indirect superheaters for various degrees of superheat, see "Stirling," published by the Stirling Boiler Company, pp. 92–96.

129. Performance of Superheaters. — Published tests of both directly and indirectly fired superheaters cover such a wide range of conditions of installation and operation that general conclusions cannot be drawn, but it may be of interest to note briefly the performances in a few specific cases.

The curves in Figs. 131, 132, and 133 are plotted from tests of a Babcock and Wilcox boiler, with 5000 square feet of water-heating surface, equipped with superheating coils of 1000 square feet area, as illustrated in Fig. 97. The furnace with ordinary short ignition arch was provided with chain grate of 75 square feet area.

Fig. 131 shows the relation between degrees of superheating and total horse power of boiler and superheater.

Fig. 132 shows the relation between horse power produced in the boiler and the percentage of boiler horse power produced in the superheater.

Fig. 133 shows the relation between the degree of superheat obtained and the horse power developed in the superheater.

Tables 37 to 39 are taken from the report of Otto Berner ("Zeit. d. Ver. Deut. Eng." and reprinted in Power, August, 1904).

Table 37 compares the heat efficiency of a steam plant equipped with directly and with separately fired superheaters, the former showing a much higher efficiency.

Table 38 compares different boilers with and without flue superheaters, showing the effect upon the temperature of the flue gases. The gain in heat efficiency of the entire plant due to the use of the superheater is decisive in each case.

Table 39 shows the gain in heat efficiency due to the use of superheaters in a number of plants equipped with fire-tube boilers.

Table 40 gives the results of tests on one of the return tubular boilers at the Spring Creek Pumping Station of the Brooklyn Waterworks (Feb. 9, 1904) with and without a superheater. The superheater, of the Foster type, was installed between the rear wall of the setting and the tube sheet.

Although the results in Tables 37 to 40 represent practice of eight years ago, they agree substantially with current practice (1912).

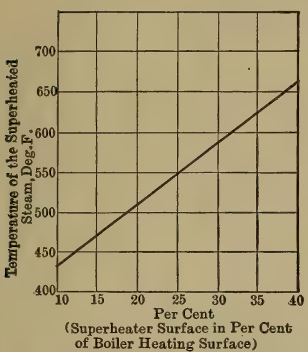


FIG. 130. Relation between Superheat and Boiler Heating Surface.

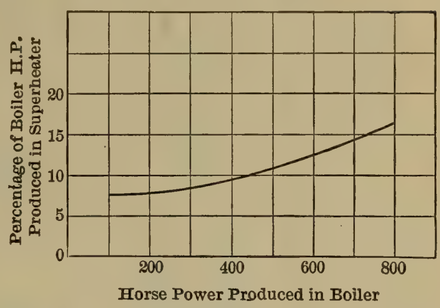


FIG. 132. Ratio of Horse Power produced in the Superheater to that developed in the Boiler.

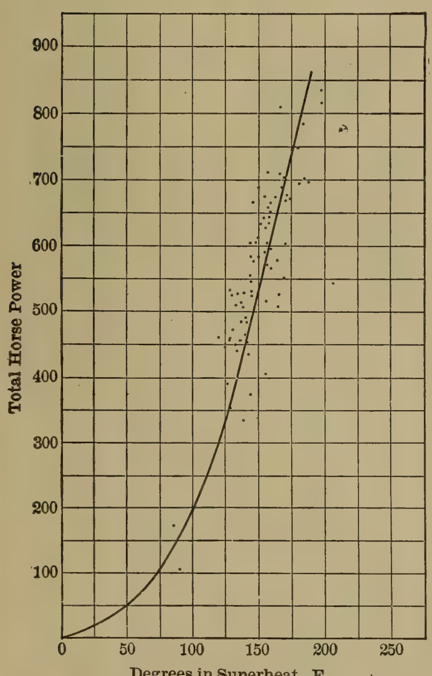


FIG. 131. Relation of Degree of Superheat to Total Horse Power developed.

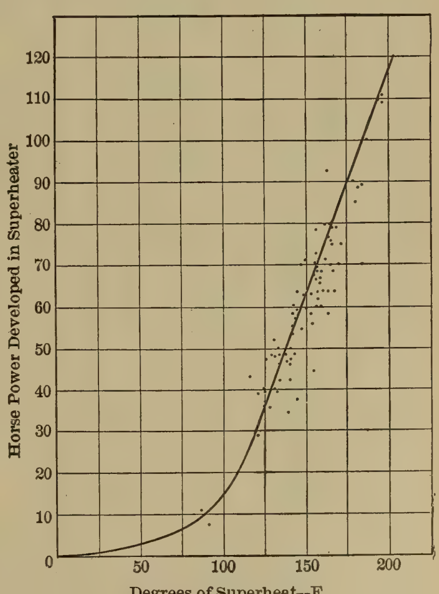


FIG. 133. Relation of Degree of Superheat to Horse Power of Superheater.

TABLE 37.

DIFFERENCE IN HEAT EFFICIENCY OF SUPERHEATERS INSTALLED IN FLUE AND SEPARATELY FIRED SUPERHEATER.
(Otto Berner, *Power*, August, 1904.)

Style of Superheater	Separately Heated Superheater. (System Uhler.)		Boiler-Flue Superheater. (Patent Schwoerer.)	
Number of test	1	2	3	4
Heating surface of superheater, Square feet	1,636.17	1,636.17	2,744.90	1,829.93
Number of boilers used in test	3	2	3	2
Boiler-heating surface, Square feet	4,843.94	3,229.29	4,843.94	3,229.29
Grate area, Square feet	104.28	69.52	104.28	69.52
Feed water per square foot of boiler-heating surface. Pounds per hour	1.96	2.75	2.04	2.8
Coal consumed per square foot of grate area, Pounds per hour	9.31	14.05	9.96	14.48
Boiler pressure, absolute, Pounds per square inch	177.48	175.64	179.91	176.35
Feed-water temperature, Degrees F.	64.76	62.6	50.00	50.00
Steam temperature on leaving superheater, Degrees F.	483.8	508.46	512.6	550.4
Heat generated per pound of coal: Boiler, B.T.U.	11,553.39	11,187.46	....	....
Heat generated per pound of coal: Superheater, B.T.U.	4,876.81	5,544.76	....	....
Heat generated per pound of coal: Entire plant, B.T.U.	10,880.53	10,605.30	12,392.33	12,229.88
Heat efficiency when using separately fired superheater less than when using boiler only, Per cent	5.70	5.33	....	....
Heat efficiency when using superheater installed in flues greater than when using boiler only, Per cent	....	....	6.87	8.75
Heat efficiency when using flue superheater greater than when using separately fired superheater, Per cent	....	....	**13.88**	**15.72**

TABLE 38.

DECREASE IN TEMPERATURE OF GASES OF COMBUSTION DUE TO SUPERHEATER INSTALLED IN FLUE.

(Otto Berner, *Power*, August, 1904).

Test	1		2		3		4		5		6	
Style of boiler	Water-Tube Boilers.											
Boiler-heating surface . . . Square feet	3,584.51		2,884.83		2,529.61		1,955.86		3,089.35		936.49	
Superheater surface . . . Square feet	....		344.45		645.85		Schwoerer Patent		1,054.9		182.99	
Grate area . . . Square feet	81.8		51.02		55.97		45.21		64.58		26.47	
Running of boiler	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.
Feed-water consumption per square foot of heating surface . . . Pounds per hour	2.46	2.23	2.68	2.66	2.91	2.69	2.31	2.10	2.47	2.03	3.70	3.50
Coal consumption per square foot of grate area . . . Pounds per hour	14.23	13.55	19.57	19.74	27.84	26.00	15.7	14.02	14.54	11.79	26.32	18.51
Boiler pressure (absolute)	130.8	131.51	149.6	148.38	182.68	181.75	122.99	121.73	149.33	150.75	149.33	151.31
Feed-water temperature . . . Degrees F.	107.6	104.72	43.97	45.68	89.9	89.6	53.24	55.4	101.3	109.04	32	67.1
Temperature of gases of combustion . . Degrees F.	**554**	**462.2**	**500**	**465.8**	**654.98**	**597.2**	**492.8**	**442.4**	**590**	**471.2**	**793.4**	**550.4**
Steam temp. on leaving superheater . . Degrees F.	......	424.94	......	486.5	......	562.6	......	489.7	......	536	......	486.5
Efficiency of boiler plant . . . Per cent	......	......	......	......	**55.9**	**59.1**	**55.63**	**60.18**	**71.9**	**78.6**	**49.3**	**62.5**
Increase in heat efficiency of plant due to use of superheated steam . . . Per cent	......	**2.3**	......	**3.3**	......	**5.4**	......	**7.6**	......	8.5	......	**21.2**
Decrease in grate area requirements due to use of superheated steam . . . Per cent	......	**4.8**	......	**0.8**	......	**6.6**	......	**10.7**	......	**18.9**	......	**23.7**
Decrease in heating surface requirements due to use of superheated steam . . . Per cent	......	9.4	......	0.9	......	9.3	......	8.8	......	17.4	......	5.5

TABLE 39.

INCREASE IN HEAT EFFICIENCY OF THE BOILER DUE TO SUPERHEATER.

(Otto Berner, *Power*, August, 1904.)

	Boiler without Superheater.	Boiler with Superheater.	Boiler without Superheater.	Boiler with Superheater.	Boiler without Superheater.	Boiler with Superheater.	Boiler with out Superheater.	Boiler with Superheater.	Boiler without Superheater,	Boiler with Superheater.
Test........................	1	2	3	4	5	6	7	8	9	10
Boiler pressure (absolute).... Lb. per sq. in.	77.21	77.21	71.25	71.25	72.53	72.53	72.24	72.24	86.18	86.18
Feed-water temperature, Deg.F.	145.4	150.8	145.4	150.8	161.6	154.4	149.0	142.6	161.6	156.2
Steam temperature.....Deg. F.	....	361.4	....	363.2	....	372.2	....	366.8	....	372.2
Evaporation per square foot of heating surface. Lb. per hour	8.56	9.12	7.3	8.1	9.01	9.95	7.23	8.56	7.7	9.09
Coal consumption per square foot of grate area.......Lb. per hour	**31.78**	**30.46**	**29.08**	**29.28**	**35.07**	**35.87**	**30.1**	**30.1**	**30.76**	**30.76**
Increase in heat efficiency due to use of superheater..........Per cent	....	**12.4**	....	**11.8**	....	**11.6**	....	**18.1**	....	**18.1**

TABLE 40.

(*Engineer, U. S.*, May 1, 1904.)

	With Superheater.	Without Superheater
Time of start	12 noon, Feb. 8	11 A.M., Feb. 11
Time of finish	12 noon, Feb. 9	11 A.M., Feb. 12
Hours run	24	24
Average steam pressure	79.3 lb.	79.4 lb.
Average water pressure, triple expansion, head in feet	0.99	1.05
Average water pressure, compound, head in feet	7.10	7.10
Average vacuum of suction for triple and compound, inches of mercury	22.90	23.21
Total head on triple, feet of water	29.05	29.46
Total head on compound, feet of water	33.04	33.39
Total double strokes, triple	30,557	34,114
Total double strokes, compound	35,395	32,158
Gallons pumped from piston displacement, total, triple	2,854,023	3,186,247
Gallons pumped from piston displacement, total, compound	2,930,706	2,662,682
Gallons pumped from piston displacement, total, triple combined	5,784,720	5,848,930
Gallons, total, pumped as measured by weir	4,492,680	4,549,480
Per cent slip	22.3	22.2
Foot pounds, weir	1,163,815,819	1,184,983,596
Total coal consumed	5,015 lb.	6,410 lb.
Per cent refuse	23.7	18.7
Total refuse	1,188	1,203
Total feed water	38,399	50,960
Duty per 100 pounds coal	23,206,696	18,486,483
Duty per 1,000 pounds steam	30,308,498	23,253,213

Per cent increase of work per 100 pounds coal....................25.5

Per cent increase of work per 1,000 pounds steam.................30.2

Per cent saving in coal per foot pound work......................20.2

Per cent saving in feed water per foot pound work................23.2

Average temperature steam leaving superheater...................527.4 deg. F.

Average temperature steam entering superheater..................320.1 deg. F.

Average degree superheat ....207.3 deg. F.

CHAPTER VI.

COAL AND ASH-HANDLING APPARATUS.

130. General. — The cost of coal and its delivery into the furnace are usually the largest items in the operating charges; hence large central stations are located, when practicable, adjacent to a railway line or water front, to minimize the cost of handling coal and ashes. Isolated stations in the business districts of large cities are usually unfavorably situated, so that the cost of handling coal and ashes is a large percentage of the total fuel cost. In large stations the amount of fuel and ash handled frequently warrants the expense of elaborate conveyor systems which would not be justified in smaller plants. In whatever way coal is supplied provision should be made for storing a quantity sufficient to operate the plant for some time in case the supply is interrupted, thereby guarding against an enforced shut-down.

If adjacent to a railway line, a side track must be provided for switching the cars. As bottom-dumping cars cannot be depended upon, provision should be made for unloading by hand or by grab bucket. If coal is delivered by water, clam-shell drop buckets are ordinarily used for unloading the barges. If the power house is located at some distance from the railroad or water the coal is generally hauled by teams in two- to five-ton loads.

131. Coal Storage. — In small stations the storage bins or coal bunkers may usually be located within the building, but in larger plants the quantity of coal consumed daily is frequently such that an immense space would be required to furnish storage capacity for even a short period of time. For example, one of the large central stations in Chicago burns an average of 60 tons of Illinois screenings per hour throughout the year. Allowing 45 cubic feet to the ton this would necessitate a space of $45 \times 60 \times 24 = 69{,}600$ cubic feet to store coal for one day's operation. A ten-days' run would require a coal pile 50 feet wide, 30 feet high, and 464 feet long. It is a good plan, if the location and character of the plant permit, to carry four or five days' supply within the plant and provide a separate building for the coal reserve. Such provision is made in the power plant of the New York Edison Company, which has a storage capacity of 150,000 tons in addition to that of the overhead bunkers.

Exposed coal piles are objectionable, because of freezing in winter, the crust sometimes freezing so hard as to necessitate the use of dynamite to break it; moreover, a slow depreciation in heat value takes place, especially with bituminous coal. This depreciation is more rapid in warm weather and in the tropics. Stored coal is oftentimes subject to spontaneous combustion, particularly when there is a large content of iron pyrites. Storage under water minimizes spontaneous combustion and depreciation in heat value. (Consult references below.)

Coal bunkers or hoppers are ordinarily placed on the same level with the boiler-room floor or above the boiler setting. The former is the cheaper as far as first cost is concerned, but necessitates additional handling of the fuel before it can be fed to the stokers. In the overhead system the coal gravitates to the stoker through down spouts. Overhead bunkers are usually found where real estate is costly. They are generally constructed of steel plates lined with concrete or of reënforced concrete. The bottoms slope at an angle of 35 to 45 degrees and empty into the coal chutes or down spouts. Fig. 136 shows the general appearance of a single overhead bunker. In some bunkers the floors are made with very slight slopes, but it is not advisable to use a slope less than the angle of repose of the coal, as it may be necessary to shovel the coal over the spouts. Convenience in framing makes the 45-degree slope the more desirable. Separate bunkers for each boiler are preferred to continuous bunkers, since fire in the coal is more readily prevented from spreading. In the new power house of Swift & Co., Chicago, Ill., the bunkers are of circular cross section instead of rectangular, as is the usual practice. The capacity of the cylindrical hopper is considerably less than that of a rectangular hopper of the same width, but is much cheaper to construct.

Ash bins are invariably lined with concrete or brickwork, since the corrosive action of the ashes would soon destroy the bare iron, and are usually located alongside the coal hopper, as in Fig. 136, so that they may be discharged by gravity. The angle of repose of most ashes is approximately 40 degrees, but the 45-degree angle is preferred on account of convenience in construction. Fig. 102 illustrates a "non-arching" type of ash hopper in which the sides are flared sufficiently to prevent the ash from packing when the bottom valves are opened.

Coal Storage Under Water: Elec. Wld., Oct. 7, 1911, p. 885, Eng. News, Dec. 24, 1908.

Calorific Value of Weathered Coals: Bulletin No. 17, Univ. of Ill., Aug. 26, 1907; Eng. News, Jan. 11, 1912, p. 64.

Spontaneous Combustion of Coal: Jl. Ind. and Chem. Eng., Mar., 1911.

Suspended Coal Bins: Power, Apr. 23, 1912, p. 602.

132. Coal Conveyors. — Coal is carried to the stokers in a variety of ways, depending upon the location of the plant, the type of stokers, and the personal tastes of the builder. Of the various methods the following are the most common:

1. Hand shoveling from coal pile to furnace.
2. Wheelbarrow or hand car and shovel.
3. Bucket conveyor.
4. Belt conveyor.
5. Hoist and hand cars.
6. Hoist and automatic-cable cars.
7. Hoist and trolley.
8. Spiral or screw conveyor.
9. Combinations of the above.

For a series of papers on conveying machinery with data pertaining to the cost of operation see Trans. A. S. M. E., Vol. 30, 1908, and Industrial Engineering, Oct., 1911, to March, 1912 (Serial).

133. Hand Shoveling. — Where possible, the coal is dumped direct from the cars or wagons into bins located in front of the boilers. In such instances one man may handle the coal and ashes and attend to the water level of 200 horse power of boilers equipped with common hand-fired furnaces. With stoking and dumping grates 300 horse power may be controlled by one man and from 800 to 1000 horse power with chain-grate stokers. This refers, of course, to average good coal not too high in ash nor productive of much clinker. Sometimes the coal cannot be stored in front of the boilers but must be hauled by wheelbarrow, cart, or rail car. For distances over 100 feet and quantities over 20 tons per day the cost of handling the coal in this way may justify the installation of an automatic conveyor system. Hand-fired furnaces and manual handling of coal and ashes are usually associated with small plants of 500 horse power and under, but a number of large stations are operated in this way with apparent economy. A notable example is the new (1907) steam power plant of the Wood Worsted Mill, Lawrence, Mass., in which 40 return tubular boilers are fired by hand. A tipcart with a capacity of one ton brings the coal a distance of 100 to 200 feet to the firing floor, and firemen shovel it on to the grate. Four men are stationed at the coal pile. One man drives two carts (one of which is being filled while the other is gone with its load), sixteen firemen attend to the furnaces, and two men dispose of the ashes. Most large plants, however, are equipped with conveying machinery, not so much because of the possible reduction in cost of operation, taking into consideration all charges fixed and operating, as

because of the large and often unreliable labor staff which it dispenses with. Hand shoveling is sometimes necessary even with modern unloading devices on account of the freezing of coal in the cars. This is particularly true of washed coals, and it is not unusual to have an entire car load solidly frozen. In this case it has to be picked and shoveled by hand, or the unloading tracks must be equipped with steam pipes and outfits for thawing purposes. A good man is capable of shoveling 40 to 50 tons of coal in eight hours when unloading a car, provided it is only necessary to shovel the coal overboard.

134. Bucket Conveyors. — One of the most common methods of automatically handling the coal from car to bunker is by means of an endless chain of traveling buckets. Many of the largest central stations in this country are equipped with such systems. The details of operation are best illustrated by a few examples.

Fig. 134 gives a diagrammatic arrangement of the link-belt overlapping pivoted bucket carrier, and Fig. 136 illustrates its application to a typical boiler plant. Coal is discharged from the railway cars into a track hopper and from there delivered by a "feeding apron" into a crusher which reduces it to such a size as can be conveniently handled by the stokers. It is then discharged into a short bucket conveyor, which carries it to the main system of buckets, and it is elevated to the proper level and discharged into the overhead bunkers. The discharge is effected by special tripping devices which engage the buckets and turn them over. The ashes are dumped from the ash pit through a series of chutes into the lower run of buckets, by which they are elevated and discharged into the ash hopper alongside the coal bunkers. From the ash hopper the ashes discharge by gravity directly into the railway cars below. The system is operated by means of two motors, one driving the crusher and the other the main bucket system. The buckets are made of either sheet steel or malleable iron.

In Fig. 134 the coal is fed to the crusher by the "reciprocating feeder," which is usually placed directly under the track hopper. The feeder consists of a heavy steel plate mounted on rollers and having a reciprocating movement effected by a crank mechanism from the carrier. The amount of coal delivered depends upon the distance the plate moves, and this can be varied by changing the throw of the eccentric. The number of strokes corresponds to the number of buckets. Any size coal can be readily handled. When the distance from track hopper to carrier is so great that the reciprocating feeder is not practicable a continuous or "belt" feeder is used to supply the crusher with fuel. The "equalizing gear" is designed to impart a pulsating motion to the driving sprocket wheel which will counteract the natural pulsation to

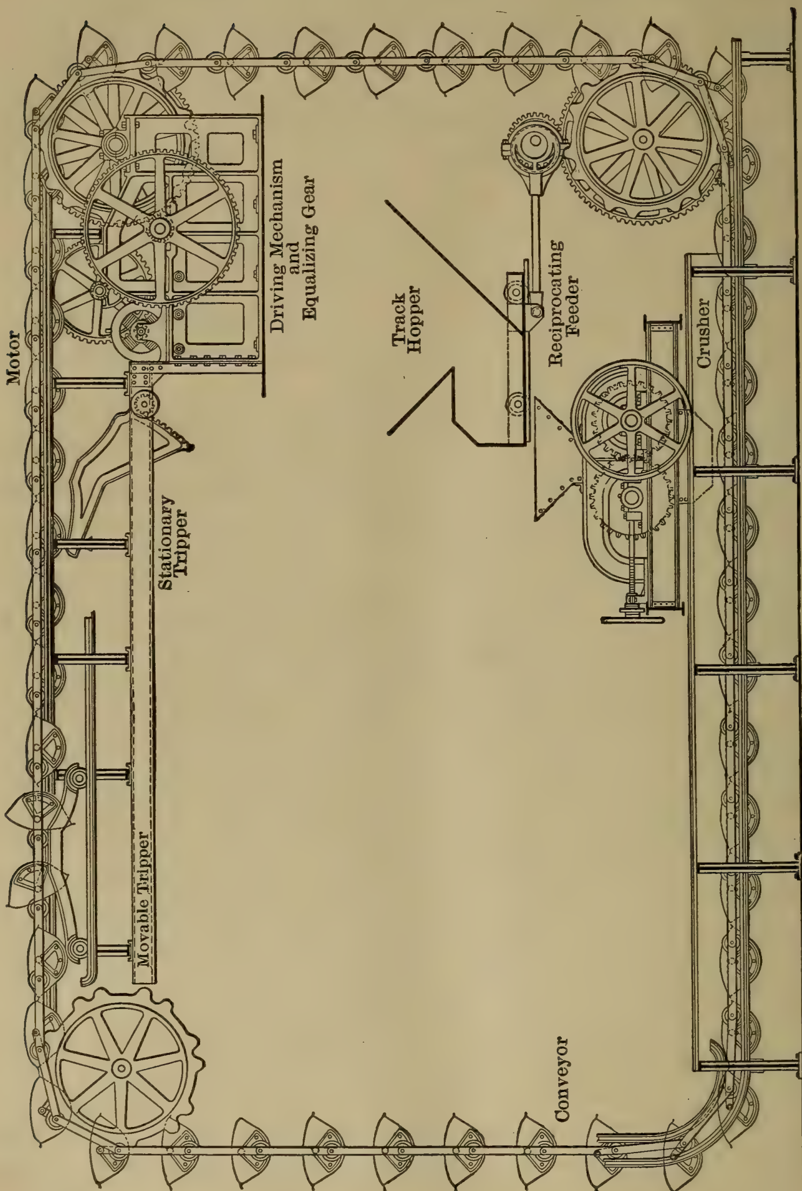


Fig. 134. Link-belt Coal-handling Apparatus.

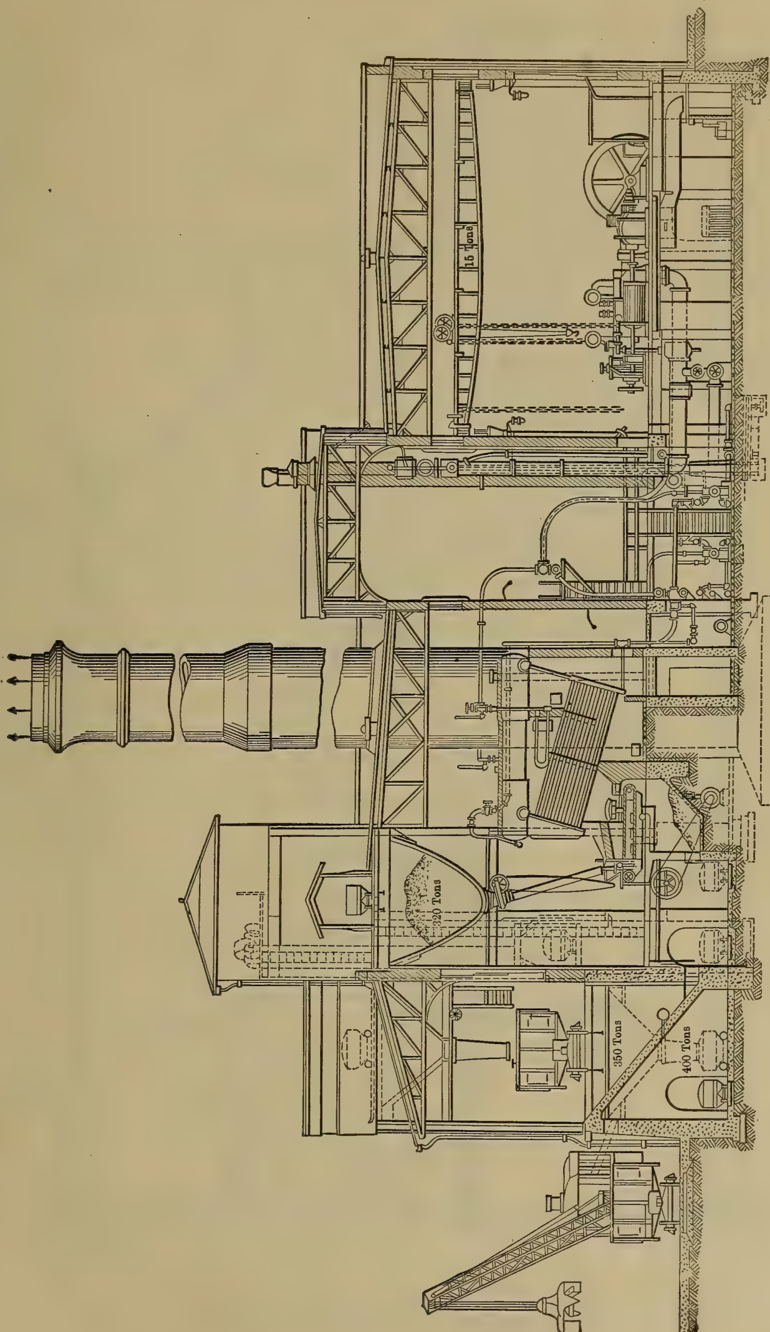


FIG. 135. Coal and Ash-handling System of the Rock Island R. R. Power Plant at Springfield, Mo.

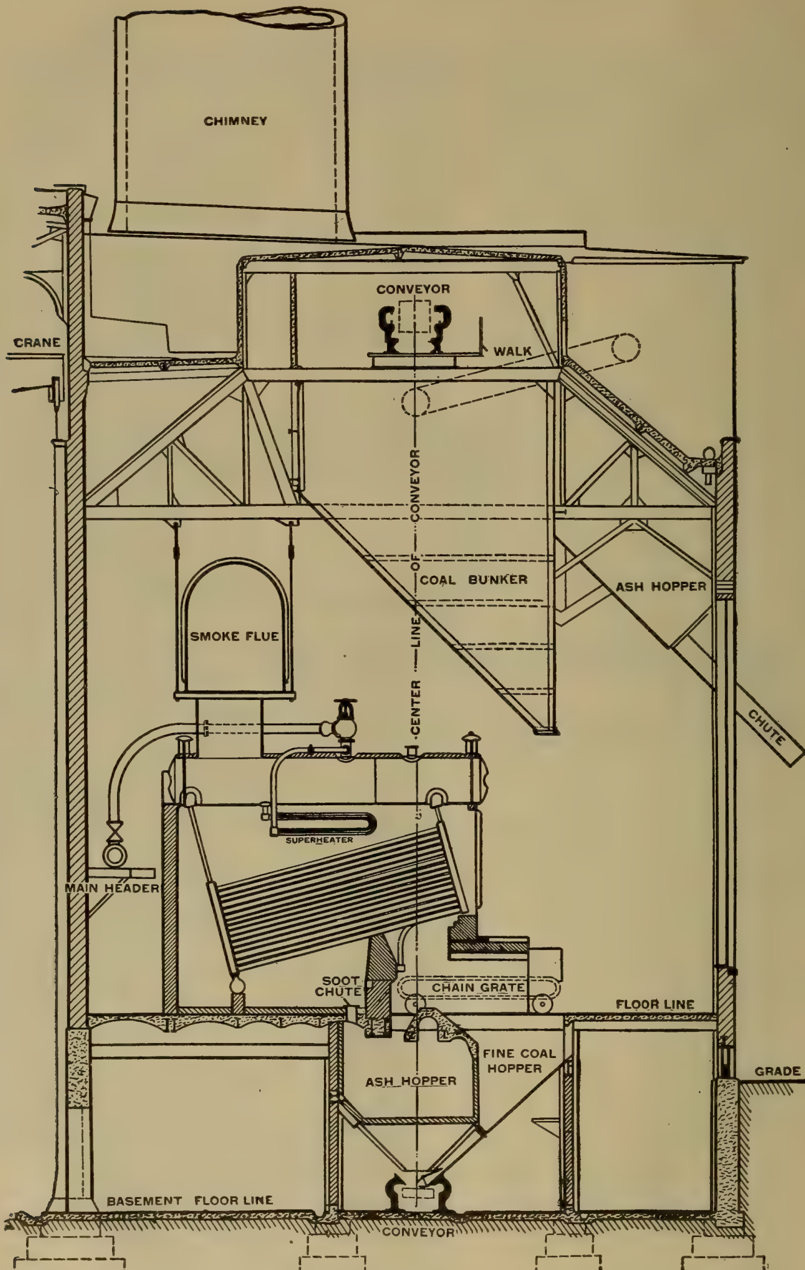


FIG. 136. Coal and Ash-handling System in the Power House of the South Side Elevated Railway Company, Chicago.

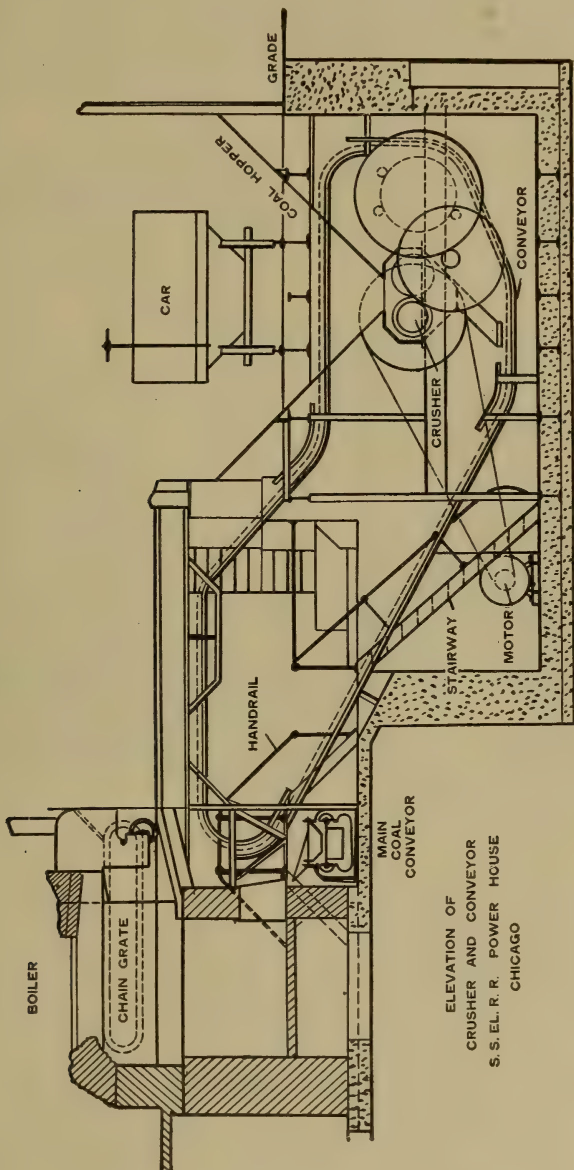


Fig. 137. Crusher and Conveyor of S. S. Elevated Railway.

which long pitch chains are subject, producing violent increase of the normal strain at frequent intervals. This is accomplished by driving the spur wheel with an eccentric pinion, causing the pitch line to describe a series of undulations corresponding to the number of sprockets on the chain wheel. Figs. 136 and 137 show the general arrangement of crusher and "cross conveyor" in the old portion of the South Side Elevated Power House, Chicago.

A coal and ash system similar to the one illustrated in Fig. 136 for a plant consisting of eight 350-horse-power boilers will cost in the neighborhood of $8000, completely installed. This does not include the cost of coal and ash bunkers.

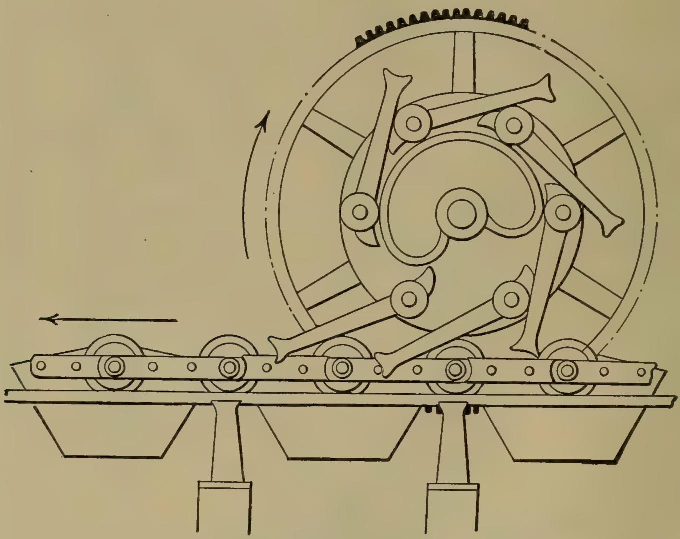


Fig. 138. Driving Mechanism of Hunt Conveyor.

The Hunt conveyor, Fig. 138, while usually called a "bucket" conveyor, is in fact a series of cars connected by a chain, each having a body hung on pivots and kept in an upright position by gravity. The chain is driven by pawls instead of by sprocket wheels. The "buckets" are upright in all positions of the chain, consequently the chain can be driven in any direction. The change of direction of the chain is accomplished by guiding the carriers over curved tracks. The chain moves slowly, and the capacity is governed by the size of the buckets. The ordinary size buckets carry two cubic feet of coal and move at a rate of fifteen buckets a minute, carrying about 40 tons per hour. Two methods of filling the buckets are employed, the "measuring" and the "spout filler." In the former each bucket is separately filled with a

predetermined amount by a suitable "measuring feeder." In the latter the material is spouted in a continuous stream, necessitating the use of overlapping buckets to prevent spilling of the material. Fig. 139 shows an application of the Hunt system to the power plant of the Rhode Island Suburban Railway, Providence, R. I.

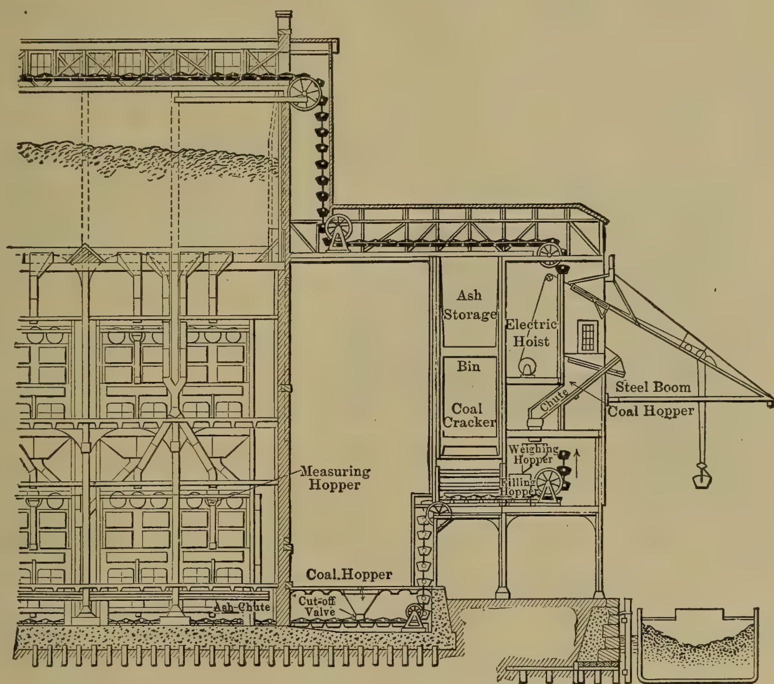


FIG. 139. Coal and Ash-handling System, Rhode Island Power House.

Fig. 140 gives a sectional elevation of the coal and ash-handling machinery at the power plant of the Commercial National Bank Building, Chicago. Underneath the sidewalk on the Clark Street side of the building is a coal-storage bin of 600 tons' capacity, served with a bucket conveyor. One leg of the conveyor reaches down to a level below the track of the Illinois Tunnel Company. By this arrangement coal can be delivered either by cars in the tunnel or by wagons from the street. In taking coal from storage a gate at the lower extremity of the hopper is opened and the coal filling the buckets is elevated and tripped into any one of the screw conveyors leading from bucket conveyor to boiler hopper. The ashes are shoveled from the ash pits into cars running in a cross tunnel under the boiler floor, and by these cars are transferred to

a dump at one side of the boiler room and discharged into Illinois Tunnel Company's cars for removal.

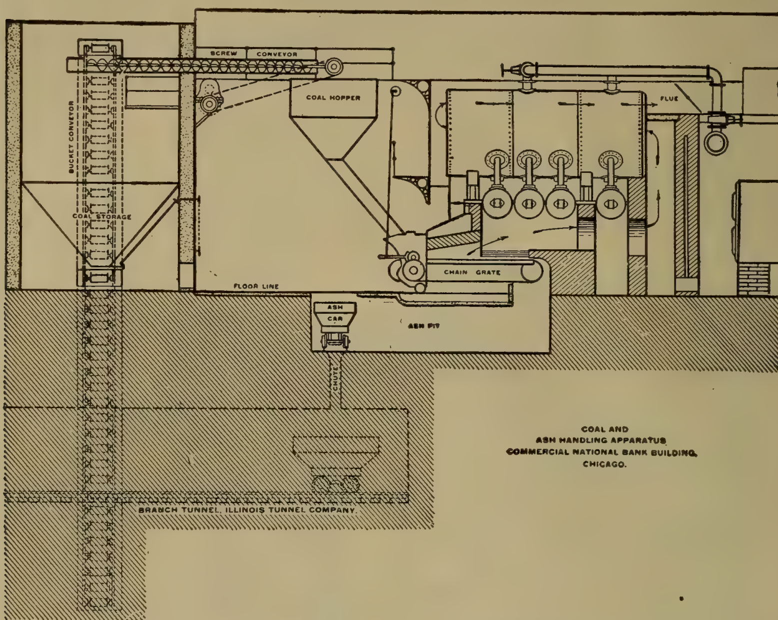


Fig. 140. Bucket and Screw Conveyor at Commercial National Bank Building, Chicago, Illinois.

135. Belt Conveyors. — The Robins belt conveyor, Fig. 141, consists essentially of a thick belt of the required width driven by suitable pulleys and carried upon idlers so arranged that the belt becomes trough-shaped in cross section. In the later designs five pulleys are employed instead of three as illustrated in order that the line of contact may more nearly approach the arc of a circle. The belt is constructed of woven cotton duck covered with a special rubber compound on the carrying side. The rubber is thicker at the middle than at the edges, since the wear is greatest in a line along the center, but the thickness of the belt is uniform throughout its entire width. The edges are reënforced with extra plies of duck to increase the tensile strength. The idlers are carried by iron or wooden framework, and are spaced from 3 to 6 feet between centers on the troughing side, according to the width of belt and the weight of the load. On the return side these distances range from 8 to 12 feet. High-speed rotary brushes with interchangeable steel bristles prevent wet, sticky material from clinging to the belt. Automatic tripping devices placed at the

proper points cause the material to be discharged where it is needed. The trippers consist essentially of two pulleys, one above and slightly in advance of the other, the belt running over the upper and under the lower one, the course of the belt resembling the letter S. The material is discharged into chutes on the first downward turn of the belt. The trippers may be movable or fixed, single or in series. Movable trippers are used when it is desired to discharge the load evenly along the entire length, as, for instance, in a continuous row of bins, while fixed trippers are employed where the load is to be discharged at certain and somewhat separated points. The movable trippers are made in two forms, "hand-driven" and "automatic." In the former they are moved from point to point by means of a hand crank. The "automatic" tripper is propelled by the conveying belt through the medium of gearing. It reverses its direction automatically at either end of the run and travels back and forth continuously distributing its load. It can be stopped, reversed, or made stationary at will. The most notable installations of this system are at the Hudson and Manhattan Railway Company's power house, Jersey City; L Street Station, Edison Illuminating Company of Boston, and the South Boston Power Station of the Boston Elevated Company.

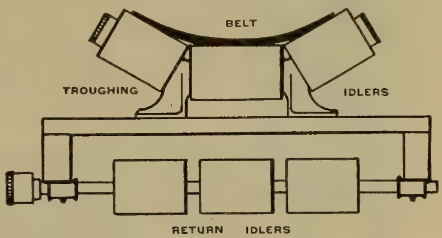


FIG. 141. Guide Pulleys, Robins Belt Conveyor.

136. Elevating Tower, Hand-car Distribution. — Fig. 142 illustrates the coal and ash-handling installation at the Aurora and Elgin Interurban Railroad power house, Batavia, Ill. Coal is delivered to the plant by railroad cars which dump directly into coal hoppers located inside a steel structure running the entire length of the building and spanned by two railroad tracks. There are 18 hoppers constructed of 17-inch brick walls fitted with steel-plate bottoms. Subdividing the storage space in this manner makes it possible to carry different grades of coal, prevents the spreading of fire, and affords a simple construction for the support of the railroad tracks. The basement of the boiler room extends underneath the hoppers, and two lines of narrow-gauge tracks are embedded in the concrete floor. Turntables at the center facilitate the switching of cars to the elevators which rise through the boiler room close to the chimney. The cars, of one ton capacity each, are of special construction, with roller-bearing axles and a combined ratchet lift and friction dump. The filled cars are pushed from un-

derneath the hoppers to two elevators which lift them to the line of tracks supported overhead across the boiler fronts. They are then pushed to the hoppers suspended above the boiler setting and the coal is dumped. These hoppers have a capacity of six tons each. From

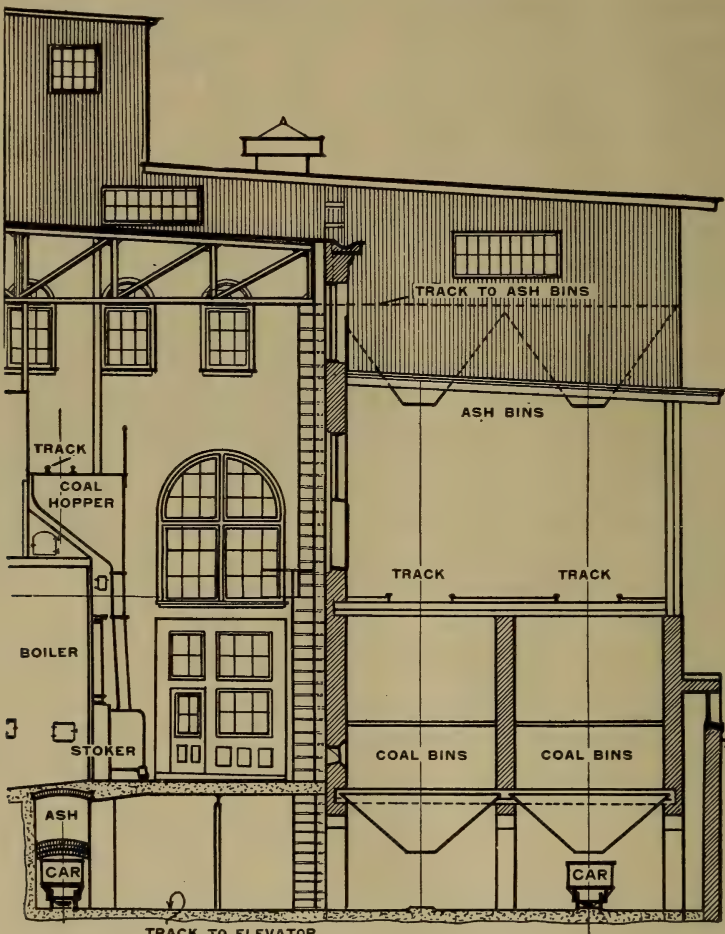


FIG. 142. Coal and Ash-handling System at the Power House of the Aurora and Elgin Interurban Railway, Batavia, Ill.

the hoppers the coal is fed to the stoker by an ordinary down spout. The ashes fall from the stokers into an ash pit, from which they may be discharged into ash cars. The ash cars are elevated to a set of tracks running at right angles to the main tracks, and are transferred to ash bins located directly over the coal bins. Coal and ashes are weighed in the small cars. There are ten boilers in this plant and four men are

required to handle the coal and ashes. The entire coal and ash-handling system cost about $10,000, and the cost of handling the coal and ashes is approximately 4 cents per ton. This does not include wages of firemen or water tenders. For a description of recent changes made in this plant see Elec. Ry. Jour., Apr. 12, 1911, p. 268.

137. Overhead Storage, Bucket Hoist. — Fig. 143 gives a general view of the coal-handling plant of the Depot Street power house of the Cincinnati Traction Company. This installation is a good example of an application of the "overhead storage gravity feed" system to an existing plant without interfering in any way with its operation. The

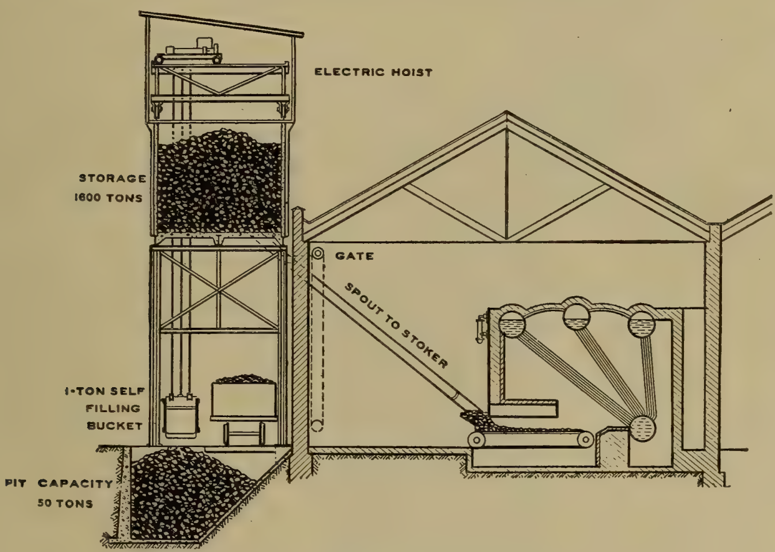


FIG. 143. Coal and Ash-handling System at the Depot Street Power House of the Cincinnati Traction Company.

system consists essentially of a receiving pit below the car tracks from which the coal is hoisted to a series of overhead bins. The coal storage is outside the boiler house in an independent structure. The bins are of steel framework with concrete floors, and are sufficiently elevated to spout coal easily to the stoker magazine. The total capacity of the overhead bins is about 1600 tons. The four bins or receiving pits have a capacity of 50 tons each, or approximately one car load, and are so situated that all four may be filled simultaneously without shifting the train. The coal-handling apparatus consists of a one-ton self-filling bucket operated on a three-motor electric crane running on rails at the top of the storage bins. The coal is hoisted from the receiving pit through suitable shafts in the bin structure and dumped into the over-

head hoppers. The maximum capacity of the hoist is 50 tons per hour. The labor required to handle the coal from car to bins is performed by one man working five hours per day and an assistant engaged a small part of the time to dump cars, clean hoppers, etc. The average daily coal consumption is approximately 200 tons. The total cost of the equipment was about $18,000 for the bins complete and $4500 for the coal-handling crane. The cost of handling the coal and ashes is approximately 1.5 cents per ton of coal. Including all charges fixed and operating the total cost of handling the coal is about 3.5 cents per ton. This does not include wages of firemen or water tenders.

138. Elevating Tower, Cable-car Distribution. — The coal and ash-handling system of the new turbine power plant of the Detroit Edison Company is a typical example of a large station equipped with elevating tower and cable-car distributers instead of the usual bucket conveyor. The system consists essentially of a lofty steel tower in which are housed at various levels a track receiving hopper, crushing rolls and feeders, weighing hopper, hoisting apparatus, etc., and a small cable railway for delivery to the bunkers. The railroad coal cars enter the tower on an elevated trestle 18 feet above grade, below which is a track receiving hopper. A two-ton "tub hoist" is filled with coal from the bottom of the receiving hopper and elevated to a 20-ton bin at the top, 120 feet above ground level. This bin has a grille bottom at one side and under the outlet a heavy duty coal crusher, thus allowing the fine coal to screen through directly while all the larger lumps are automatically delivered to the crusher. From the two bins the small cable cars are filled for dumping into the desired bunkers over the boiler rooms. The cars are arranged for automatic dumping by means of adjustable trips which may be located at any point. The entire system has a capacity of from 125 to 150 tons of coal per hour and is motor-driven. The ash-handling system consists of brick-lined concrete hoppers underneath each pair of stokers which discharge their contents by gravity into the small cars operated on the track system in the boiler-house basement.

When handling 600 tons per day of 24 hours the cost of operation is approximately 20 cents per ton from coal car to ash car. This includes wages of firemen and water tenders.

139. Hoist and Trolley. — Fig. 144 illustrates a very simple and economical method of handling coal and ashes as installed by the Jeffrey Manufacturing Company at the power plant of the Scioto Traction Company. If the coal car is of the dump type the contents are discharged directly into the coal pit from which the coal is removed by grab bucket and transferred either to the overhead bunker

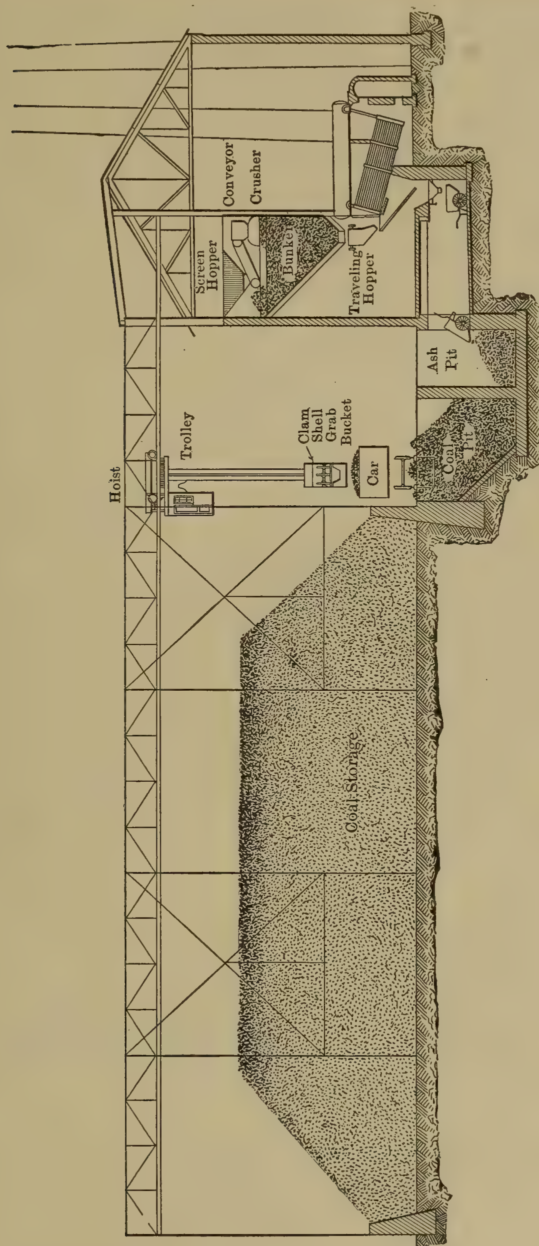


FIG. 144. Coal and Ash-handling System at the Power House of the Scioto Valley Traction Co.

or to the storage pile. If the coal car is of the gondola type the coal is removed directly from the car by the grab bucket. The bucket is hoisted and carried on the trolley into the building over the screen hoppers where it discharges its contents; the finer particles fall directly into the bunker and the larger lumps are automatically delivered to the crusher. The grab bucket will take about 98 per cent of the coal in the car, leaving only 2 per cent to be handled by hand. Coal is fed to the stokers by means of a traveling electric hopper which receives its supply from the overhead bunkers. The present capacity of the plant is 50 tons per hour taken from the car or pit to stock pile.

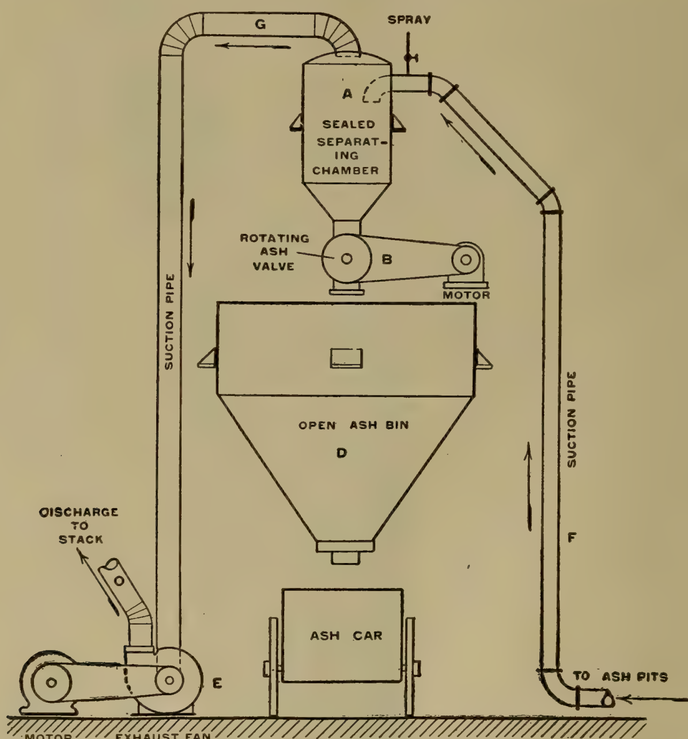


FIG. 145. Diagrammatic Arrangement of the "Vacuum" Ash-handling System.

140. "Vacuum" Ash Conveyor. — Fig. 145 gives a diagrammatic arrangement of a recently patented ash-conveying system depending upon the velocity of a column of air for moving the ashes. The system is simple in operation and low in first cost. One end of special cast-iron header *F* leads to the ash pits of the various boilers by means of branch tubes, and the other end is connected with a sealed separating

chamber A. Each branch pipe is fitted with simple circular openings directly underneath each ash-pit door for admitting ashes and which are kept covered except when in operation. Exhauster E creates a partial vacuum in chamber A and draws in air at a high velocity from the opening in the ends of the branch pipes. Ashes raked into the pipes through the openings are caught by the rapidly moving column of air and forced into chamber A. The ashes fall to the bottom and are fed into the main ash pit by a slowly revolving ash valve B. Air and dust are withdrawn from the top of the separator chamber through pipe G and discharged to the stack or to waste. A spray is introduced into pipe F to reduce dust. The process is a continuous one and the ashes may be completely removed from the ash bin without interfering with the operation of the exhauster. In a later construction the ash bin and separating chamber are included in one chamber, thus doing away with the revolving ash valve and the small motor operating it. In this latter design the bin is never completely empty, a certain depth of ashes being maintained to seal the bottom at all times.

At the Armour Glue Works, Chicago, Ill., this system is applied to a boiler plant of thirteen boilers, aggregating 4800 horse power, and cost, completely installed, $5600. As originally installed the separating chamber had a volume of about 35 cubic feet and the suction intake was placed 58 feet above the ash-pit level. The revolving ash valve made about 13 r.p.m., and was driven by a one-horse-power motor. In the present installation the separating chamber and motor-operated ash valve are dispensed with and the discharge pipes lead directly into the main ash bin, which has a capacity of 60,000 pounds of wet ashes and is constructed of five-sixteenths-inch sheet iron. The exhauster (a 30-foot Root blower) has a capacity of about 8000 cubic feet per minute at 265 r.p.m., and is driven by a 75-horse-power motor. Under normal conditions of operation the motor requires 50 horse power when delivering 250 pounds of ash per minute, and the vacuum on the suction side of the exhauster is 3.3 inches of mercury. The pipe from the ash bins to the separating chamber is 10 inches in diameter and is constructed of extra heavy chilled cast-iron pipe. The piping from the separating chamber to exhauster and to stack is 22 inches in diameter and is constructed of number 16 and number 20 galvanized iron. The ashes are raked by hand from the ash pits to the suction openings of the branch pipes, and are handled dry, the dust being taken along with the ashes. Elbows are soon worn out by the abrasive action of the ashes, and tees are used instead, since the accumulation in the "dead" end receives the impact and takes up the wear. The cost of handling the ashes in this installation is approximately 7 cents per ton.

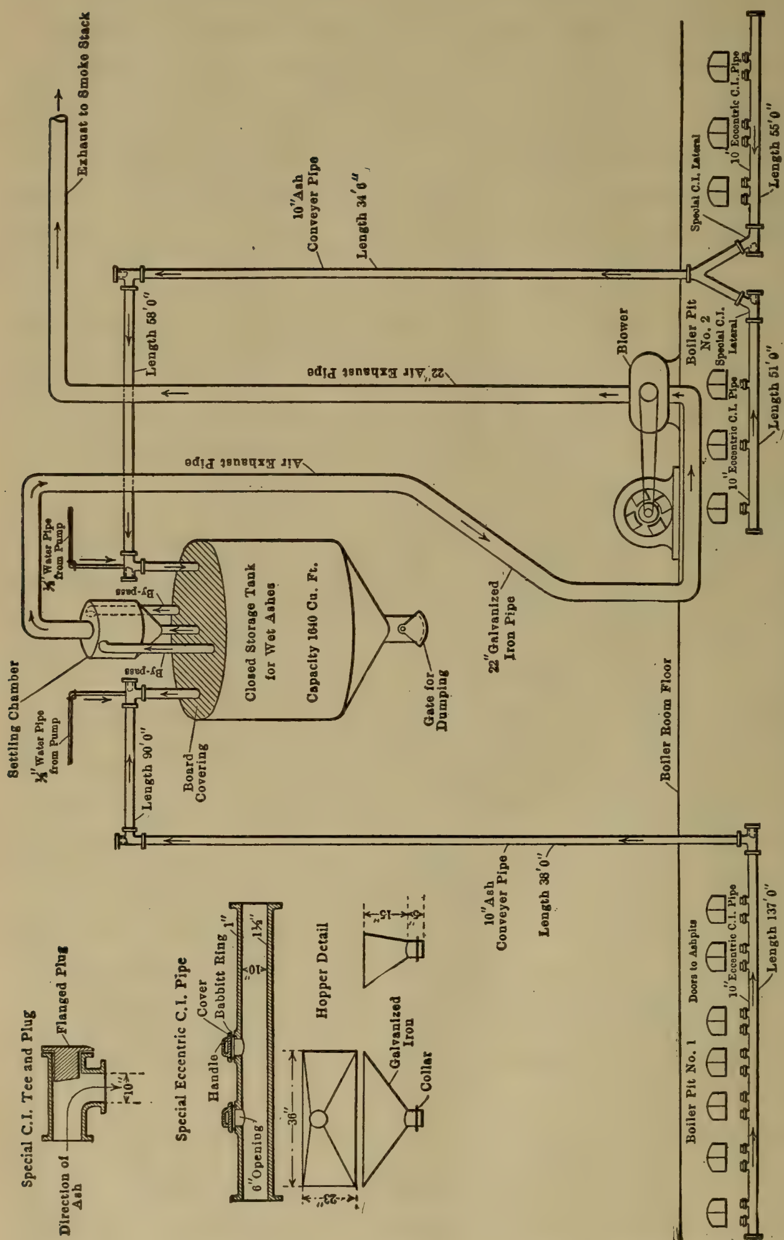


FIG. 146. Diagrammatic Arrangement of Vacuum Ash-Handling System at the Armour Glue Works, Chicago, Ill.

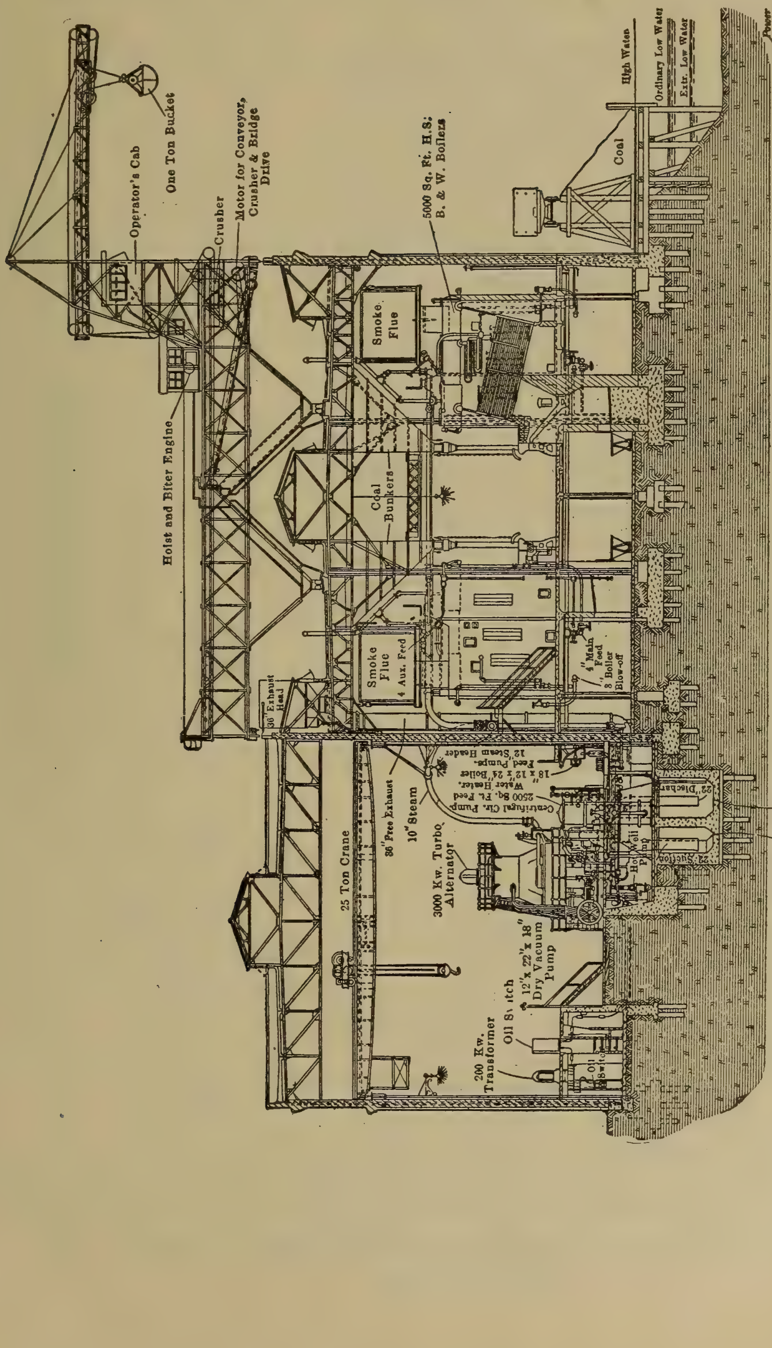


FIG. 147. Coal and Ash-Handling System. Norfolk Traction Co., Norfolk, Va.

141. Cost of Handling Coal and Ashes. — In large stations where a number of men are employed to handle coal and ashes only it is a simple matter to divide the cost of handling into the various stages, thus:

1. Cost of unloading cars or barges.
2. Cost of conveying coal to bunkers.
3. Cost of feeding coal to furnace.
4. Cost of removing ashes.

These costs are usually expressed in cents or dollars per ton of coal burned, or in terms of cents or dollars per horse-power hour or kilowatt hour of main prime mover output. Item number 3 is oftentimes included under "boiler-room attendance" and items 1, 3, and 4 under "coal and ash handling." Not infrequently all four items are included under "attendance." So much depends upon the character of stokers and furnace, size of boilers, and the like, that general figures on the cost of handling the coal and ashes are of little value unless accompanied by a description of the equipment. For the sake of general comparison the most satisfactory method of expressing the cost is in dollars per ton of coal from coal car to ash car. This includes wages of coal and ash passers, repair men, and boiler tenders. In small stations the coal and ash handling is done by the boiler tenders, in which case it is impracticable to separate the items mentioned above, and the cost is ordinarily included under attendance. An average figure for handling coal by barrow and shovel is not far from 1.6 cents per ton per yard up to the distance of five yards, then about 0.1 cent per ton per yard for each additional yard. With automatic conveyors the operating cost, not including wages of firemen and water tenders, varies with the size of plant and the type of conveyor, and ranges anywhere from a fraction of a cent per ton to four or five cents per ton. The larger the plant and the greater the amount of coal handled the lower will be the cost per ton. In comparing the relative costs of manual and automatic handling, fixed charges of at least 15 per cent of the first cost of the mechanical equipment should be charged against the latter in addition to the cost of operation. In large central stations equipped with stokers and conveyors and consuming 200 tons or more of coal in twenty-four hours, the cost of handling the coal from coal car to ash car, including wages of firemen and water tenders, will range between 10 cents and 18 cents a ton.

142. Coal Hoppers. — Fig. 148 shows a front and side elevation of a typical set of stationary weighing hoppers as applied to the boilers of the Quincy Point power plant of the Old Colony Street Railway Company, Quincy Point, Mass. Each battery of boilers is provided

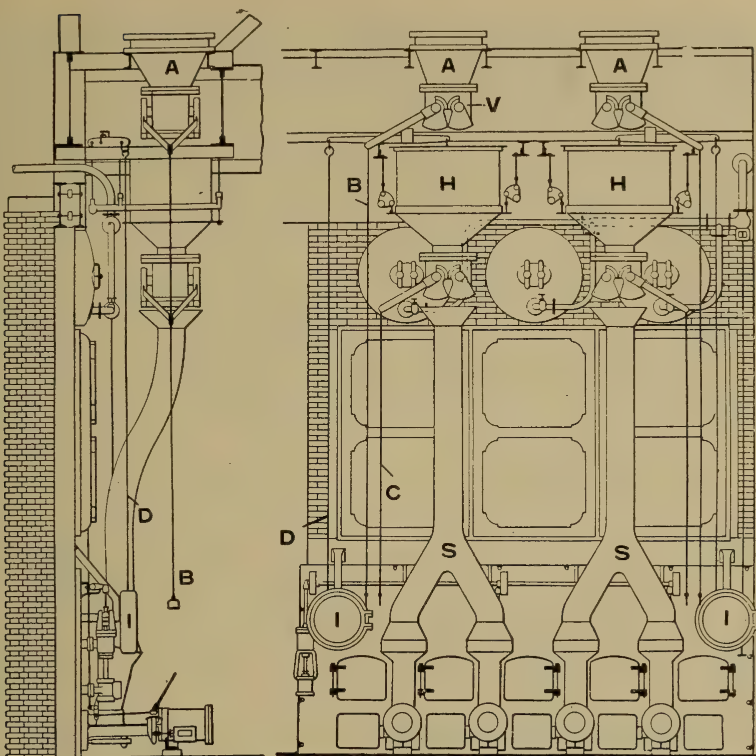


FIG. 148. Stationary Coal Weighing Hoppers.

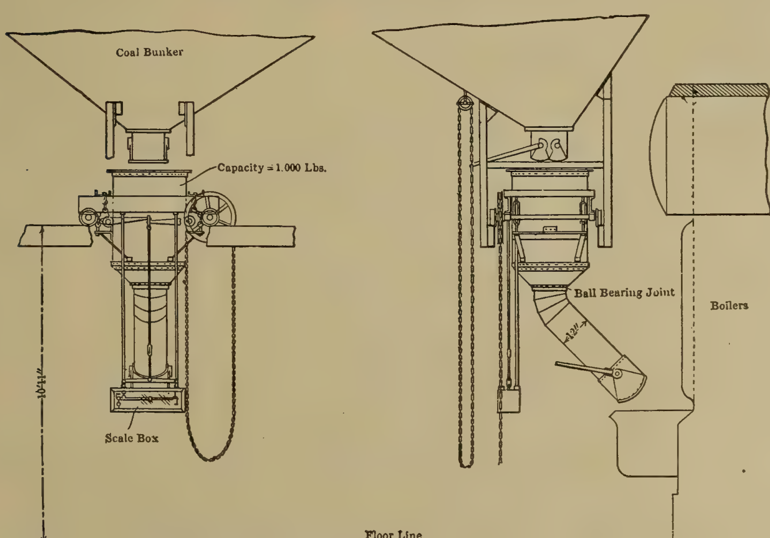


FIG. 149. Traveling Coal Hoppers.

with an independent set of hoppers. The bottoms of the overhead coal bunkers lead into the small hoppers A, A. The operation of any single weighing hopper is as follows: Coal is fed from the overhead bunkers to weighing hopper H by means of valve V. The weight of coal in the weighing hopper is transmitted by a system of levers and knife edges to the inclosed scale beam I and noted in the usual way. The weighed charge of coal is then admitted to the down spout S by means of valves similar to those at V.

Although separate weighing hoppers for each battery, as illustrated in Fig. 148, offer many advantages, they are quite costly and it is not unusual to install one or more large weighing hoppers mounted on overhead traveling carriages so that one may supply a number of boilers (Fig. 149). At the Armour Glue Works, Chicago, the coal

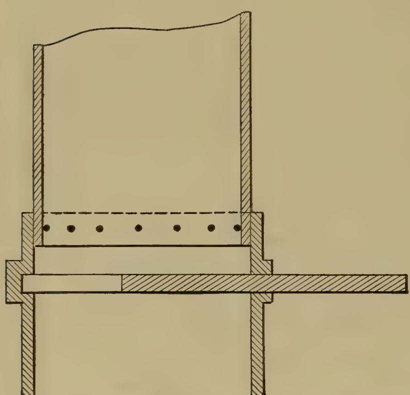


Fig. 150. Common Slide Coal Valve.

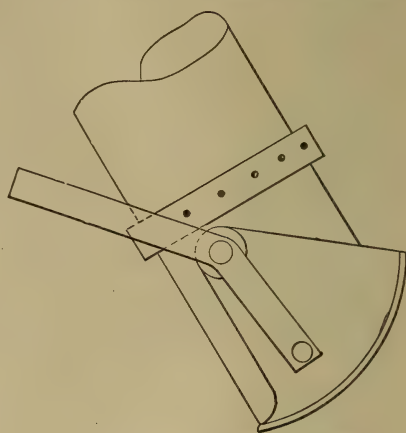


Fig. 151. Simplex Coal Valve.

supply is stored in one large overhead bunker of 1000 tons' capacity. A five-ton motor-driven traveling hopper receives its supply from this central bunker and delivers it to the various boilers. One man operates the traveling hopper, tends to the coal valves, and supplies all boilers with coal.

Weighing hoppers are sometimes made automatic; that is, the opening and closing of valves, feeding of coal, and recording of weight are automatically performed by the weight of the coal itself. The scale is set for discharges of a certain weight and continues to discharge this amount automatically. In the few plants which are equipped with automatic weighing hoppers the capacity of the hopper is approximately 100 pounds per discharge. These hoppers are necessarily more complicated and more costly than the ordinary weighing hoppers, and it is a question whether the advantages offset the extra first cost and main-

tenance charges. A small automatic hopper of 100 pounds discharge capacity costs approximately $400 as against $250 for the ordinary weighing device. For a description of a coal meter see paragraph 429

143. Coal Valves. — Figs. 150 to 154 illustrate the principles of a few well-known coal valves. They may be conveniently grouped into two classes according to the location of the coal pocket: (1) those drawing the coal from overhead bunkers and (2) those drawing from the side of a bin. In the first class come the simple *slide valve* and the *simplex* and *duplex rotating* valve. In the latter are the *flap* valve and the *rotating* valve. They are made in various sizes and designs, but those illustrated are examples of the most common types. The simple slide valve, Fig. 150, is applicable only to small size coal and to small spouts, since coarse or lump coal may get in the way and prevent proper closing. The simplex valve, Fig. 151, consists of a rotating jaw actuated by a lever. There are no rubbing surfaces, and the jaws cut through the material without jamming. The duplex valve, Fig. 152, consists of two rotating jaws connected to a common actuating lever. The jaws move simultaneously, so that even a partially open valve delivers the coal centrally. When closing the valve the flow is gradually stopped by the decreasing width of the opening and there is but little resistance to the movement of the jaws. The largest valve can easily be operated by hand.

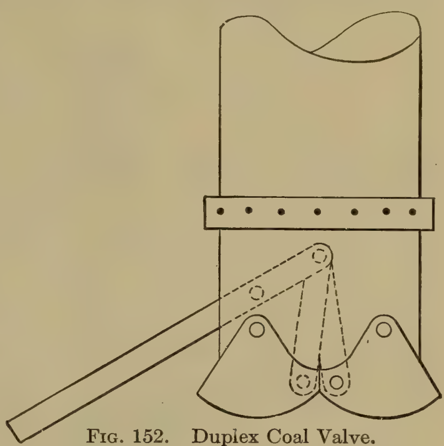


FIG. 152. Duplex Coal Valve.

The flap valve, Fig. 153, is the simplest form for drawing coal from a side bin. It consists merely of an iron flap hinged to the bottom of the chute. The valve is lowered to let the coal run over its top and is raised to stop the flow. It cannot be clogged or get jammed in closing. The flap is raised and lowered by a simple lever.

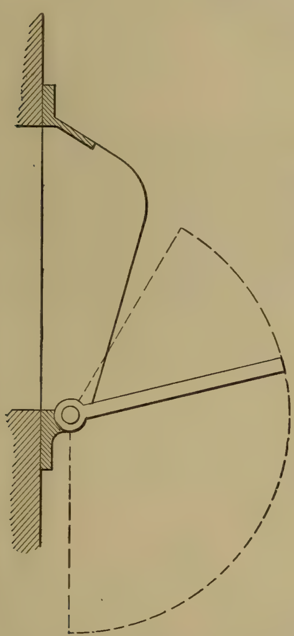


FIG. 153. Common "Flap" Coal Valve.

For very large bins, where the valves are to be opened and closed frequently, the "Seaton" valve, Fig. 154, is usually preferred. This valve consists of two jaws EE', and TT' pivoted to suitable framework at O and actuated by lever A. The valve is shown fully closed. Raising

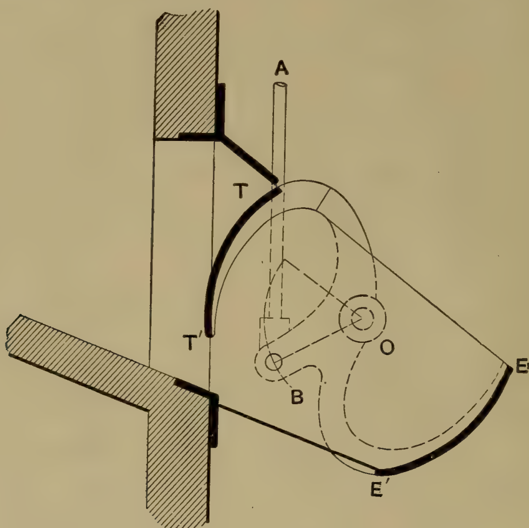


FIG. 154. "Seaton" Coal Valve.

lever A causes the cut-off blade EE' to rotate about O and permits the coal to flow through the space between the edge of the jaw E and the end of the chute. The rate of flow is regulated by the width of this opening. The cut-off blade does not reach a stop, hence there is no possibility of a lump of coal getting in the way and preventing the prompt closing of the valve.

CHAPTER VII.

CHIMNEYS.

144. Chimney Draft. — Draft produced by a chimney depends upon so many conditions and involves such a large number of variables that empirical methods of proportioning, based upon actual performances, are more to be relied upon than theoretical calculations. Draft is due to the difference in the weight of the column of hot light gases in the stack and that of the cooler and heavier surrounding atmosphere, the latter tending to flow into the base and thereby force the lighter gases out the top of the stack. The commonly accepted theory of chimney draft is based upon Peclet's hypothesis that the flow through the furnace flues and chimney may be represented by the equation

$$h = \frac{u^2}{64.4}\left(l + G + \frac{fl}{m}\right),$$

in which

h = the head of fluid producing the flow, feet;

u = velocity of the gases in the chimney, feet per second;

G = a coefficient to represent the resistance to the passage of air through the coal;

l = total length of the path of the gases, feet;

m = area of cross section divided by the perimeter;

f = a coefficient depending upon the nature of the surfaces over which the gases pass.

From experiments on chimneys and boilers Peclet gives in connection with this theory the following values of coefficients G and f:

$$G = 12, \qquad f = 0.012,$$

on the basis of 20 to 24 pounds of coal burned per square foot of grate surface per hour. On account of the variation in practice of the factors u, f, and G and the difficulty of determining them engineers prefer to use the modified formulas given further on.

The theoretical difference of pressure or *intensity of draft* may be determined as follows:

Let H = height of chimney in feet;
T = absolute temperature of the freezing point, degrees F.;
T_1 = absolute temperature of the gases in the chimney;
T_2 = absolute temperature of the outside air;
P = average atmospheric pressure = 14.7 pounds per square inch;
P_2 = observed atmospheric pressure;
W = weight of a cubic foot of air at 32 degrees F. and pressure P;
W_1 = weight of a cubic foot of chimney gas at 32 degrees F. and pressure P.

Then the weight of a cubic foot of hot gas in the chimney will be

$$W_1 \frac{P_2}{P} \cdot \frac{T}{T_1}, \tag{51}$$

and the weight of a cubic foot of cold air outside will be

$$W \frac{P_2}{P} \cdot \frac{T}{T_2}. \tag{52}$$

The weight of a column of hot gas H feet high and one foot square (assuming uniform density and temperature) will be

$$W_1 H \frac{P_2}{P} \cdot \frac{T}{T_1}. \tag{53}$$

Similarly, the weight of the cold-air column will be

$$WH \frac{P_2}{P} \cdot \frac{T}{T_2}, \tag{54}$$

and the difference in pressure or the intensity of draft will be

$$D = H \frac{P_2}{P}\left(\frac{WT}{T_2} - \frac{W_1 T}{T_1}\right), \tag{55}$$

where D is in pounds per square foot.

By making $P = P_2 = 14.7$, $T = 492$, $W = 0.0807$, $W_1 = 0.084$, and D_1 = pressure in inches of water ($D_1 = 0.192\,D$), equation (55) assumes the familiar form

$$D_1 = H\left(\frac{7.64}{T_2} - \frac{7.95}{T_1}\right). \tag{56}$$

By assuming $W = W_1 = 0.081$ and $P = 14.7$ equation (55) may be written

$$D_1 = 0.52\,HP_2\left(\frac{1}{T_2} - \frac{1}{T_1}\right). \tag{57}$$

This latter form is ordinarily used where the atmospheric pressure differs considerably from that at sea level, as at high altitudes. Table 41 gives the density of air and chimney gases at various temperatures.

Example: Required the maximum theoretical draft obtainable from a chimney 150 feet high, atmospheric pressure 14.7 pounds per square inch, temperature of outside air 60 degrees F., temperature of chimney gases 550 degrees F.

Here $H = 150$, $T_2 = 460 + 60 = 520$, $T_1 = 460 + 550 = 1010$.

Substituting these values in equation (55),

$$D_1 = 150\left(\frac{7.64}{520} - \frac{7.95}{1010}\right) = 1.02 \text{ inches of water,}$$

which is about 20 per cent greater than the draft actually obtained, and represents the maximum possible under the given conditions, neglecting the resistance offered by the chimney and the pressure

TABLE 41.

DENSITY AND SPECIFIC VOLUME OF AIR AND CHIMNEY GASES AT VARIOUS TEMPERATURES.

Air.				Chimney Gases.					
t	*s*	*v*	*d*	*t*	*d*	*t*	*d*	*t*	*d*
0	11.581	.935	.086353	200	.06334	430	.04695	660	.03730
5	11.706	.945	.085424	210	.06239	440	.04643	670	.03697
10	11.832	.955	.084513	220	.06147	450	.04592	680	.03665
15	11.931	.965	.083623	230	.06058	460	.04542	690	.03633
20	12.085	.976	.082750	240	.05971	470	.04493	700	.03602
25	12.211	.986	.081895	250	.05887	480	.04445	710	.03571
30	12.337	.996	.081058	260	.05805	490	.04398	720	.03540
32	12.387	1.000	.080728	270	.05726	500	.04353	730	.03511
35	12.463	1.006	.080238	280	.05648	510	.04308	740	.03481
40	12.589	1.016	.079434	290	.05573	520	.04264	750	.03453
45	12.715	1.026	.078646	300	.05499	530	.04221	760	.03424
50	12.841	1.037	.077874	310	.05428	540	.04178	770	.03396
55	12.967	1.047	.077117	320	.05358	550	.04137	780	.03369
60	13.093	1.057	.076374	330	.05290	560	.04096	790	.03342
62	13.144	1.061	.076081	340	.05224	570	.04056	800	.03316
65	13.220	1.067	.075645	350	.05159	580	.04017	900	.03072
70	13.346	1.077	.074930	360	.05096	590	.03979	1000	.02861
75	13.472	1.087	.074229	370	.05035	600	.03942	1100	.02678
80	13.598	1.098	.073541	380	.04975	610	.03905	1200	.02516
85	13.724	1.108	.072865	390	.04916	620	.03869	1300	.02373
90	13.851	1.118	.072201	400	.04859	630	.03833	1400	.02245
95	13.976	1.128	.071550	410	.04803	640	.03798	1500	.02131
100	14.102	1.138	.070910	420	.04749	650	.03764	1800	.01848
110	14.354	1.159	.069665	......	....	......	....	2000	.01698

d = density, pounds per cubic foot.

t = temperature, degrees F.

s = specific volume, cubic feet per pound.

v = comparative volume, volume at 32° = 1.

Density of chimney gas taken 0.085 pound per cubic foot at 32° F. and 29.92 inches of mercury.

(Rankine, "Steam Engine," gives the density at 32° F. as varying from 0.084 to 0.087.)

TABLE 42.

THEORETICAL DRAFT PRESSURE IN INCHES OF WATER. CHIMNEY 100 FEET HIGH.[1]

Temp. in the Chimney.	Temperature of the External Air — Barometer, 14.7 Pounds per Square Inch.[2]										
	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
200	.453	.419	.384	.353	.321	.292	.263	.234	.209	.182	.157
220	.488	.453	.419	.388	.355	.326	.298	.269	.244	.217	.192
240	.520	.488	.451	.421	.388	.359	.330	.301	.276	.250	.225
260	.555	.528	.484	.453	.420	.392	.363	.334	.309	.282	.257
280	.584	.549	.515	.482	.451	.422	.394	.365	.340	.313	.288
300	.611	.576	.541	.511	.478	.449	.420	.392	.367	.340	.315
320	.637	.603	.568	.538	.505	.476	.447	.419	.394	.367	.342
340	.662	.638	.593	.563	.530	.501	.472	.443	.419	.392	.367
360	.687	.653	.618	.588	.555	.526	.497	.468	.444	.417	.392
380	.710	.676	.641	.611	.578	.549	.520	.492	.467	.440	.415
400	.732	.697	.662	.632	.598	.570	.541	.513	.488	.461	.436
420	.753	.718	.684	.653	.620	.591	.563	.534	.509	.482	.457
440	.774	.739	.705	.674	.641	.612	.584	.555	.530	.503	.478
460	.793	.758	.724	.694	.660	.632	.603	.574	.549	.522	.497
480	.810	.776	.741	.710	.678	.649	.620	.591	.566	.540	.515
500	.829	.791	.760	.730	.697	.669	.639	.610	.586	.559	.534
550	.863	.828	.795	.762	.731	.700	.671	.644	.618	.593	.585
600	.908	.873	.839	.807	.776	.746	.717	.690	.663	.638	.613

1. For any other height multiply the tabular figure by $\frac{H}{100}$, where H is the height in feet.

2. For any other pressure multiply the tabular figure by $\frac{P}{14.7}$, where P is the barometric pressure in pounds per square inch.

required to impart velocity to the gases. Table 42 has been computed from formula (57) and gives the maximum theoretical draft in a chimney 100 feet high for different flue-gas temperatures.

The intensity of draft required to produce best results depends upon the kind and condition of fuel, the thickness of fire, character of grate, and resistance of the breeching, tubes, baffles, dampers, etc. As stated above, the loss of draft in the chimney proper approximates 20 per cent of the total, that in the breeching is taken as 0.1 inch per 100 feet of flue, and 0.05 inch for each right-angle bend; the loss in the boiler varies from 0.3 to 0.6 inch, depending upon the type;* the loss in the furnace varies between wide limits, and depends upon the kind of fuel and the rate of combustion. The curves in Fig. 155 compiled by the Stirling Company and published in their book "Stirling" give the furnace drafts necessary to burn various kinds of fuels at different combustion rates, and give an idea of the influence of the character of the fuel and the rate of combustion.

* Specific figures may be obtained from the manufacturers.

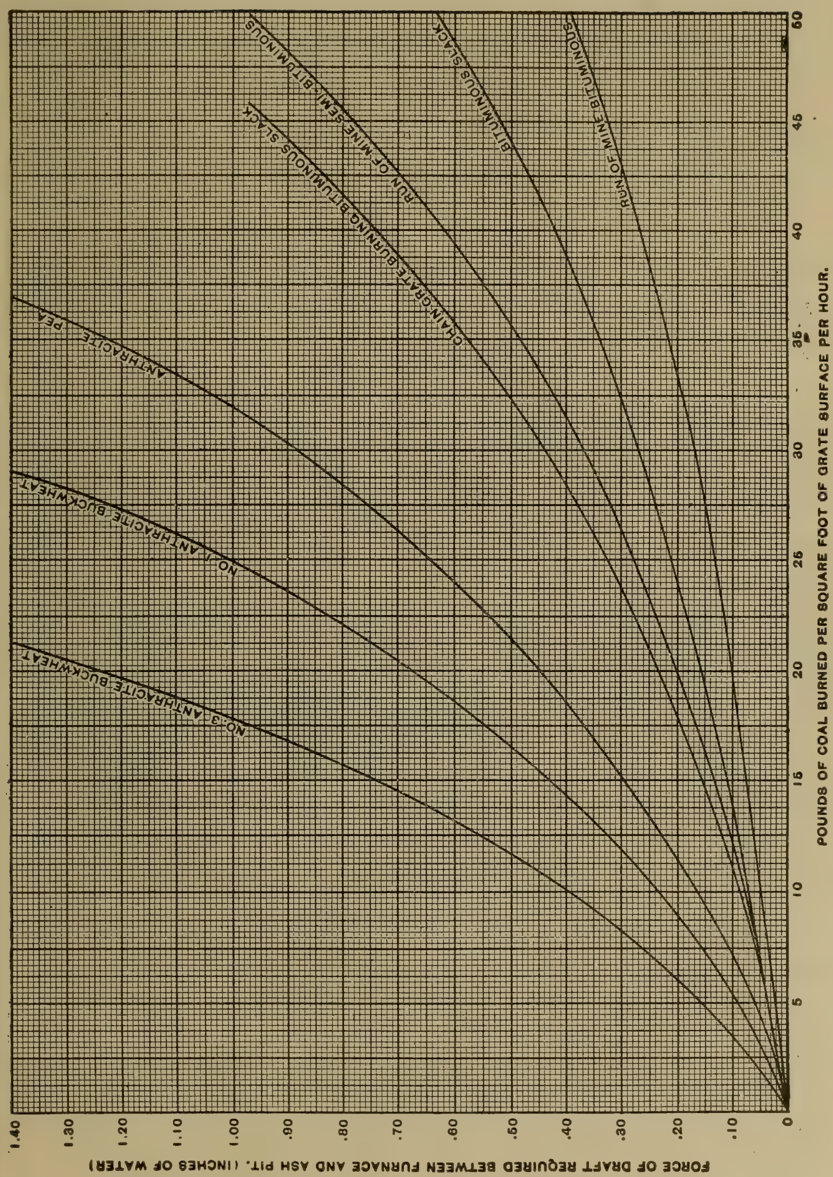


FIG. 155. Relation between Draft and Rates of Combustion.

Example: Determine the probable draft necessary to burn 30 pounds bituminous run of mine per hour per square foot of grate when the outside air is 60 degrees F., the temperature of the chimney gases 550 degrees, and the flue is 100 feet long, with two right-angle bends.

The losses will be divided approximately as follows:

	Inch.
Loss in furnace (from curves in Fig. 155)	0.17
Loss in boiler (average)	0.40
Loss in flue, 100 feet at 0.10 per 100	0.10
Loss in turns, 2 × 0.05	0.10
	0.77

Since the loss in the chimney alone approximates 20 per cent of the total, $0.77 \div 0.80 = 0.96$ will be the maximum pressure difference. From equation (56),

$$D_1 = H\left(\frac{7.64}{T_2} - \frac{7.95}{T_1}\right).$$

Substituting for the given values of D_1, T_1, and T_2 in above equation,

$$0.96 = H\left(\frac{7.64}{520} - \frac{7.95}{1010}\right),$$

from which $H = 142$, height of stack necessary to produce a draft of 0.17 inch in the furnace.

Table 43 gives the results of a test of a 100-foot unlined steel chimney, showing the variation in draft at different points in the stack.

The curves in Figs. 156 and 157 are taken from Bulletin 21, U. S. Bureau of Mines, 1911, and are of interest in illustrating the pressure drops throughout the boiler for different conditions of operation. Fig. 156 shows the pressure drops through the combustion chamber and over the fuel bed for a hand-fired Heine boiler when the total drop from ash pit to uptake is varied and the resistances to flow of gas are kept constant. Fig. 157 shows the pressure drops through the same equipment when the total drop is varied and the resistances to flow are kept constant. The curves show that the drop through any portion of the path of the gases bears a constant ratio to the total drop, provided the resistances to the flow of gases remain constant. For further data bearing out this fact consult the bulletin referred to.

Theory of Chimney Draft: National Engineer, Dec., 1911, p. 588, Jan., 1912, p. 39; Power, March, 1906, Feb., 1900, p. 12; Engr. U. S., Jan. 15, 1903, May 15, 1902, p. 313; Trans. A.S.M.E., 11–451, 762, 772, 974, 984; Bulletin No. 21, U. S. Bureau of Mines, 1911.

145. Chimney Formulas. — Rational methods of determining the height and area of chimneys being cumbersome and unwieldy and of doubtful value for practical use, the various empirical formulas out-

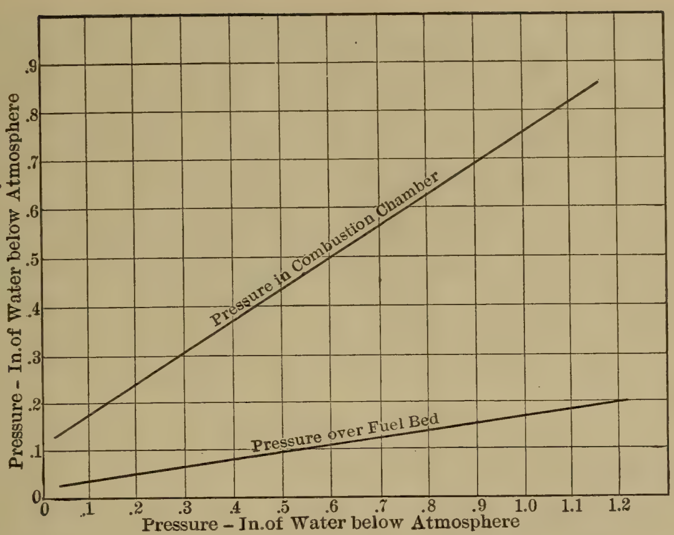


FIG. 156. Pressure Drops through Two Parts of Heine Hand-fired Furnace when Total Drop from Ash Pit to Uptake is Varied and Resistances to Flow are Kept Constant.

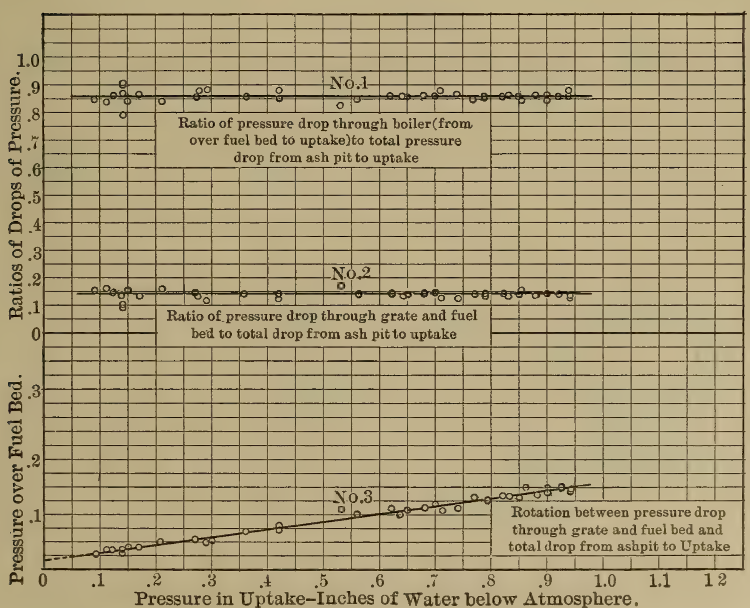


FIG. 157. Pressure Drops through Two Parts of Heine Hand-fired Furnace when the Total Drop is Varied and Resistances to Flow are Kept Constant.

lined in Table 44 are quite commonly used. They give good results within the limits of the assumptions upon which they are based, but otherwise may lead to absurd results, their applicability depending largely upon the available data covering the various losses with the particular kind, quality, and condition of coal, and conditions of operation. Occasionally practical and local considerations fix the height of the stack irrespective of theoretical deductions. The logical procedure is to *determine first the height of chimney necessary to produce the draft at the desired maximum rate of combustion* and then to proportion the area by such formulas as (2), (4), or (5), to suit the quantity of fuel to be burned.

The following heights have been found to give good results in plants of moderate size:

	Feet.
With free-burning bituminous coal	90
With anthracite, medium and large sizes	120
With slow-burning bituminous	140
With anthracite pea	150
With anthracite buckwheat	175
With anthracite slack	200

TABLE 43.

CHIMNEY DRAFT.

Test of a 100-Foot Unlined Steel Chimney 3 Feet in Diameter at Massachusetts Institute of Technology. (Peabody & Miller, "Steam Boilers," p. 121.)

	Draft, Inches of Water.		Temperature, Fahrenheit.	
	Maximum.	Minimum.	Maximum.	Minimum.
Over the grate	0.24	0.218	....	....
At the bridge wall	0.382	0.372	....	....
Half-way between bridge and back end of boiler	0.410	0.374	....	....
At the back end of boiler	0.354	0.334	....	....
In uptake near boiler	0.572	0.543	403	389
In stack 34 feet above grate	0.440	0.414	396	374
In stack 51 feet above grate	0.334	0.312	380	368
In stack 68 feet above grate	0.216	0.168	370	354
In stack 85 feet above grate	0.122	0.086	345	314

The chimney serves two 80-horse-power boilers. During test one was banked and the combustion at the grate of the working boiler was 19.8 pounds per square foot of grate surface per hour. Coal burned per hour, 590 pounds.

For plants of 800 horse power or more the height of stack should never be less than 150 feet, regardless of the kind of coal used.

Referring to Table 44, formulas (1), (2), (6), (7), and (9) are based upon a fuel consumption of 13 to 15 pounds of anthracite and 22 to 26 pounds of bituminous coal per square foot of grate area per hour. In formulas (3), (4), and (9), the diameter is dependent solely upon the quantity of coal burned per hour and the height is determined mainly by the rate of combustion per square foot of grate. The results accord well with practice. With western coals formula (3) gives results rather too large and the constant should be 120 instead of 180. Formula (5) is perhaps the most used and has met with much approval. It is based on the assumptions that

1. The draft of the chimney varies as the square root of the height.
2. The retardation of the ascending gases by friction may be considered due to a diminution of the area of the chimney or to a lining of the chimney by a layer of gas which has no velocity and the thickness of which is assumed to be 2 inches. Thus, for square chimneys,

$$E = D^2 - \frac{8D}{12} = A - \frac{2}{3}\sqrt{A}, \tag{58}$$

and for round chimneys,

$$E = \frac{\pi}{4}\left(D^2 - \frac{8D}{12}\right) = A - 0.591\sqrt{A}. \tag{59}$$

For simplifying calculations the coefficient of $\sqrt{A}$ may be taken as 0.6 for both square and round chimneys, and the formula becomes

$$E = A - 0.6\sqrt{A}. \tag{60}$$

3. The horse-power capacity varies as the effective area E.
4. A chimney should be proportioned so as to be capable of giving sufficient draft to permit the boiler to develop much more than its rated power in case of emergencies or to permit the combustion of 5 pounds of fuel per rated horse power per hour.
5. Since the power of the chimney varies directly as the effective area E and as the square root of the height H, the formula for horse power for a given size of chimney will take the form

$$\text{H.P.} = CE\sqrt{H}, \tag{61}$$

in which C is a constant, found by Mr. Kent to be 3.33, obtained by plotting the results from numerous examples in practice.

The formula then assumes the form

$$H.P. = 3.33\, E \sqrt{H}, \tag{62}$$

or

$$H.P. = 3.33\, (A - 0.6 \sqrt{A}) \sqrt{H}, \tag{63}$$

from which

$$H = \left(\frac{0.3 \text{ H.P.}}{E}\right)^2. \tag{64}$$

Table 45 has been computed from equation 5, Table 44.

Many engineers simply adopt the following proportions:

Internal area of chimney at top, one-seventh grate area for bituminous coal.

Internal area of chimney at top, one-ninth grate area for anthracite coal.

Example: Determine the area and diameter of a stack for a 2000-horse-power plant to operate under the following conditions: Rated load 2000 horse power; maximum overload 40 per cent of rated; flue 150 feet long, with one right-angle bend; average rate of combustion 20 pounds of bituminous coal per square foot of grate surface per hour; atmospheric temperature 60 degrees F.; flue-gas temperature at overload 600 degrees F.; coal burned per boiler horse power 4 pounds.

With modern types of steam engines or turbines an overload of 40 per cent has little effect on the economy of the prime mover, and the boiler efficiency is but slightly reduced, but an additional allowance of 25 per cent should be made in estimating the overload combustion rate.

The maximum rate of combustion then will be

$$20 + \left(\frac{40 + 25}{100}\right) 20 = 33,$$

pounds per square foot of grate surface per hour.

The draft required at the point where the flue enters the chimney, considering the various losses, will be found as follows:

	Inch.
Furnace (see curves, Fig. 155)	0.3
Boiler (assumed)	0.4
Flue, 150 feet at 0.1 inch per 100 feet	0.15
Turns, 1 at 0.05	0.05
	0.9

From formula (56),

$$D_1 = H\left(\frac{7.64}{T_2} - \frac{7.95}{T_1}\right).$$

TABLE 44.—CHIMNEY FORMULAS.

Index.	Author.	References.	Formulas. Horse Power.	Formulas. Height, Feet.	Area, Square Feet.
1	Adams.	Adams, "Handbook for Mechanical Engineers," p. 155.	....	$H=\left(\frac{F}{14\,A}\right)^2.$	$A=\frac{F}{14\sqrt{H}}.$
2	Christie.	Christie, "Chimney Design," p. 22.	$HP=3.24\,A\sqrt{H}.$	$H=\left(\frac{F}{KA}\right)^2.$	$A=\frac{F}{K\sqrt{H}}.$
3	Gale........	Trans. A.S.M.E., Vol. XI, p. 463.	....	$H=\frac{180}{t}\left(\frac{F}{G}\right)^2.$	$A=0.07\,F^{\frac{3}{4}}.$
4	"Ingenieurs Taschenbuch"	....	....	$H=0.216\left(\frac{F}{G}\right)^2+6\,D.$	....
5	Kent........	Trans. A.S.M.E., Vol. XII, p. 81.	$HP=3.33\,E\sqrt{H}.$	$H=\frac{\overline{0.3\,HP}^2}{E^2}.$	$E=\frac{0.3\,HP}{\sqrt{H}}.$
6	Lange........	Eng. Record, July 20, 1901, p. 52.	....	$H=15\,D+32.5.$	$A=0.00049\,BF.$
7	Molesworth...	Molesworth, "Pocket Book" ...	$HP=1.28\,A\sqrt{H}.$	$H=\left(\frac{F}{12\,A}\right)^2.$	$A=\frac{F}{12\sqrt{H}}.$
8	Nagle........	Power, November, 1902, p. 29...	$HP=2\,D^2\sqrt{H}.$	$H=\frac{\overline{HP}^2}{4\,D^4}.$	$D=\left(\frac{HP}{2\sqrt{H}}\right)^{\frac{1}{2}}.$
9	Nystrum.	Nystrum, "Mechanics," ed. 1882, p. 423.	$HP=1.45\,A\sqrt{H}.$	$H=\left(\frac{F}{12\,A}\right)^2.$	$A=\frac{HP}{1.45\sqrt{H}}.$
10	Rankine......	Christie, "Chimney Design," p. 9.	....	$H=\frac{13\,\frac{V}{2\,g}\left(\frac{w}{n\delta}\right)^2\left(\frac{T_1}{T_0}\right)^2}{0.96\,\frac{T_1}{T_2}-1-\frac{0.06\,D}{2\,gA}\left(V^2\,\frac{wT_1}{n\delta T_0}\right)^2}.$	....
11	Smith........	Smith, "Boiler Practice," p. 423.	....	$H=\left(\frac{F}{12\,A}\right)^2.$	$A=\frac{0.0825\,F}{\sqrt{H}}.$
12	Stirling......	1905 issue of "Stirling," Stirling Boiler Company.	....	$H=\frac{d'}{0.8\,M}.$	$D'=C\,(HP)^{0.4}.$

H. Height above grate; ft.
A. Inside area at top; sq. ft.
E. Effective area $=A-0.6\sqrt{A}$.
D. Inside diameter at top; ft.
HP. Boiler horse power.
F. Lbs. coal burned per hour.

G. Grate surface; sq. ft.
d'. Net available draft; in. H_2O.
$K=\begin{cases}\frac{F}{1.89\,G} & \text{for bituminous coal.}\\ \frac{F}{G} & \text{for anthracite.}\end{cases}$

$C=\begin{cases}4.68 \text{ for unlined stack.}\\ 4.92 \text{ for lined stack.}\end{cases}$
t. Temperature flue gas; deg. F.
M. Theoretical draft per foot of chimney height.
B. Weight of air per lb. of coal.
V. Cu. ft. air per lb. of coal, 32° F.

w. Lb. fuel burned per second.
n. Ratio grate to chimney area.
δ. Weight of 1 cu. ft. chimney gas, 32° F.
T_2. Temperature external air; deg. F., abs
T_1. Temperature flue gas; deg. F., abs.
T_0. = 461.
D' = Diameter in inches.

TABLE 45.

SIZE OF CHIMNEYS FOR STEAM BOILERS.

Kent's Formula.

Diam. Inches.	Area (a) Sq. Ft.	Effective Area $E=A-0.6\sqrt{A}$ Sq. Ft.	Height of Chimney.														Equivalent Sq. Chimney Side of Sq. $\sqrt{E}+4$ Inches.
			50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.	225 ft.	250 ft.	300 ft.	
			Commercial Horse Power of Boiler.*														
18	1.77	.97	23	25	27	29	......	......	......	......	......	......	......	......	......	......	16
21	2.41	1.47	35	38	41	44	......	......	......	......	......	......	......	......	......	......	19
24	3.14	2.08	49	54	58	62	66	......	......	......	......	......	......	......	......	......	22
27	3.98	2.78	65	72	78	83	88	......	......	......	......	......	......	......	......	......	24
30	4.91	3.58	84	92	100	107	113	119	......	......	......	......	......	......	......	......	27
33	5.94	4.48	......	115	125	133	141	149	156	......	......	......	......	......	......	......	30
36	7.07	5.47	......	141	152	163	173	182	191	204	......	......	......	......	......	......	32
39	8.30	6.57	......	......	183	196	208	219	229	245	......	......	......	......	......	......	35
42	9.62	7.76	......	......	216	231	245	258	271	289	316	......	......	......	......	......	38
48	12.57	10.44	......	......	......	311	330	348	365	389	426	......	......	......	......	......	43
54	15.90	13.51	......	......	......	......	427	449	472	503	551	595	......	......	......	......	48
60	19.64	16.98	......	......	......	......	536	565	593	632	692	748	......	......	......	......	54
66	23.76	20.83	......	......	......	......	......	694	728	776	849	918	981	......	......	......	59
72	28.27	25.08	......	......	......	......	......	835	876	934	1,023	1,105	1,181	1,253	......	......	64
78	33.18	29.73	......	......	......	......	......	......	1,038	1,107	1,212	1,310	1,400	1,485	1,565	......	70
84	38.48	34.76	......	......	......	......	......	......	1,214	1,294	1,418	1,531	1,637	1,736	1,830	2,005	75
90	44.18	40.19	......	......	......	......	......	......	......	1,496	1,639	1,770	1,893	2,008	2,116	2,318	80
96	50.27	46.01	......	......	......	......	......	......	......	1,712	1,876	2,027	2,167	2,298	2,423	2,654	86
102	56.75	52.23	......	......	......	......	......	......	......	1,944	2,130	2,300	2,459	2,609	2,750	3,012	91
108	63.62	58.83	......	......	......	......	......	......	......	2,090	2,399	2,592	2,771	2,939	3,098	3,393	96
114	70.88	65.83	......	......	......	......	......	......	......	......	2,685	2,900	3,100	3,288	3,466	3,797	101
120	78.54	73.22	......	......	......	......	......	......	......	......	2,986	3,226	3,448	3,657	3,855	4,223	107
132	95.03	89.18	......	......	......	......	......	......	......	......	3,637	3,929	4,200	4,455	4,696	5,144	117
144	113.10	106.72	......	......	......	......	......	......	......	......	4,352	4,701	5,026	5,331	5,618	6,155	128

* Based on a consumption of 5 pounds of fuel per boiler horse power. For any other rate multiply the tabular figure by the ratio of 5 to the maximum expected coal consumption per horse power per hour.

Substituting the following values:

$$T_2 = 60 + 460 = 520, \qquad T_1 = 600 + 460 = 1060,$$

$$D_1 = \text{maximum draft} = \frac{0.9}{0.8} = 1.12 \text{ inch},$$

$$1.12 = H\left(\frac{7.64}{520} - \frac{7.95}{1060}\right),$$

whence the necessary height of stack is

$$H = 160 \text{ feet (approximately)}.$$

Substituting the value of H in Kent's formula, the effective area is found to be

$$E = \frac{0.3 \text{ H.P.}}{\sqrt{H}} = \frac{0.3 \times 2000}{\sqrt{160}} = 47.5 \text{ square feet},$$

corresponding to an actual diameter of 93 inches.

The actual velocity of the gases in a chimney is from 0.25 to 0.35 of that theoretically possible, the latter being based on the assumption that the maximum theoretical draft or pressure difference is available for producing velocity. (See paragraph 162.) For examples and calculations from actual tests see National Engineer, Dec., 1911, p. 588, and Jan., 1912, p. 39.

146. Height of Chimneys for Boilers using Oil Fuel. — Experimental data relative to chimneys for boilers using oil fuel are rather meager and discordant, but a study of a number of recent installations seems to indicate that the area need not exceed 50 per cent of that required by the same boiler using bituminous coal. A height 80 to 100 feet above the grate usually affords sufficient draft to force the boilers 50 per cent above rating, but in a number of large installations the chimneys have been designed on the coal-burning basis so as to provide sufficient capacity in case it proves necessary at a future date to revert to the use of coal. See Jour. A.S.M.E., Oct., 1912, p. 1499.

147. Classification of Chimneys. — Chimneys may be grouped into three classes according to the material of construction:

1. Steel.
2. Reënforced concrete.
3. Masonry.

Steel chimneys have many advantages and are finding much favor in large power plants, especially where economy of space warrants the erection of the stack over the boiler, in which case the structural work of the boiler setting answers for both boiler and chimney. Among the advantages are: (1) ease and rapidity of construction; (2) less weight for a given internal diameter and height; (3) less surface

exposed to the wind; (4) lower cost; (5) smaller space required; (6) slightly higher efficiency if properly calked, for there can be no infiltration of cold air as is likely through the cracks in masonry. The chief disadvantage is the cost of keeping the stack well painted to prevent rust and the corrosive action of the sulphur in the coal.

Steel chimneys may be:

1. Guyed.
2. Self-sustained.

148. Guyed Chimneys. — Guyed sheet-iron or steel chimneys or stacks held in position by guy wires are employed in small sizes on account of their relative cheapness. They seldom exceed 52 inches in diameter and 75 feet in height. A heavy foundation is unnecessary, and the stack may be supported by the boiler breeching. The small short stacks are ordinarily riveted in the shop, ready for erection, larger sizes being shipped in sections and riveted at the place of instal-lation. The guy wires are usually fastened to an angle iron or band at about two-thirds the height, and anchored at a distance from the base equal to the height of the band above the ground.

For very tall stacks two sets of guys are used, from four to six wires being fastened to each band, and designed to withstand a wind pressure of 30 pounds per square foot of projected area of the stack. Turn-buckles are employed to equalize tautness. Table 46 gives the thick-ness of material, with approximate cost and weight, of guyed stacks of different heights and areas.

TABLE 46.

APPROXIMATE WEIGHT AND COST OF GUYED SHEET-STEEL CHIMNEYS.

Height, Feet.	Diameter, Inches.	Thickness of Shell, B.W.G.	Approximate Weight per Foot, Pounds.
40	18	16	13
45	20	16	14
45	22	14, 16	20, 15
50	24	14, 16	22, 16
50	26	14	23.5
55	28	14	25
60	30	12, 14	34, 27
65	32	12, 14	36, 28
70	34	10, 12	48, 39
75	36	10, 12	51, 41

Approximate cost per pound, 4 cents to 10 cents, including cost of sections riveted and punched, ready for assembling, the higher figure referring to the smaller stacks.

149. Self-sustaining Steel Chimneys. — Steel chimneys over 52 inches in diameter are usually self-supporting. They may be built with or without a brick lining, but the lining is preferred, since it prevents radiation and protects the inside from the corrosive action of the flue gases. Since the lining plays no part in the strength of the chimney, it is made only thick enough to support its own weight, and usually of a low-grade fire brick or carefully burned common brick or both. In average practice the fire brick extends 20 or 30 feet above the breeching, the remainder of the lining being of common brick. In chimneys up to 80 inches internal diameter, the upper course is $4\frac{1}{2}$ inches thick and increases $4\frac{1}{2}$ inches in thickness for each 30 to 40 feet to the bottom. In larger chimneys about 8 inches is the minimum thickness. The lining is generally set in contact with or close to the shell, though a space of from 1 to 2 inches is sometimes left between the brickwork and the shell to allow for expansion. This space is occasionally filled with sand.

Self-sustaining stacks may be straight or tapered, and are generally made with a flared or bell-shaped base whose diameter and length are $1\frac{1}{2}$ to 2 times the internal diameter of the stack. The base is riveted to a heavy cast-iron plate bolted to a concrete foundation of sufficient mass to insure stability.

Fig. 158 gives the details of one of the steel chimneys at the power house of the South Side Elevated Railroad, Chicago, Ill.

150. Thickness of Plates. — The sheet is thickest at the bottom, decreasing toward the top of the stack. The proper thickness for any given section may be determined by treating the shaft as a uniformly loaded cantilever, the stresses being expressed by the equation

$$Ph = S\frac{I}{e} = S\frac{\pi}{32}\left(\frac{D_1^4 - D_2^4}{D_1}\right), \tag{65}$$

in which

P = the total wind pressure in pounds,

h = length of the chimney in inches to the center of wind pressure ($h = L/2$ for a cylindrical chimney),

S = safe stress. A low value of 6000 pounds per square inch for single-riveted joints and 8000 for double-riveted joints is recommended, for the reason that a tube of such large diameter with thin walls will hardly fail by rupture according to the formula, but by flattening and bending.

$\frac{I}{e}$ = sectional modulus,

D_1 = external diameter of the shell, inches,

D_2 = internal diameter of the shell, inches.

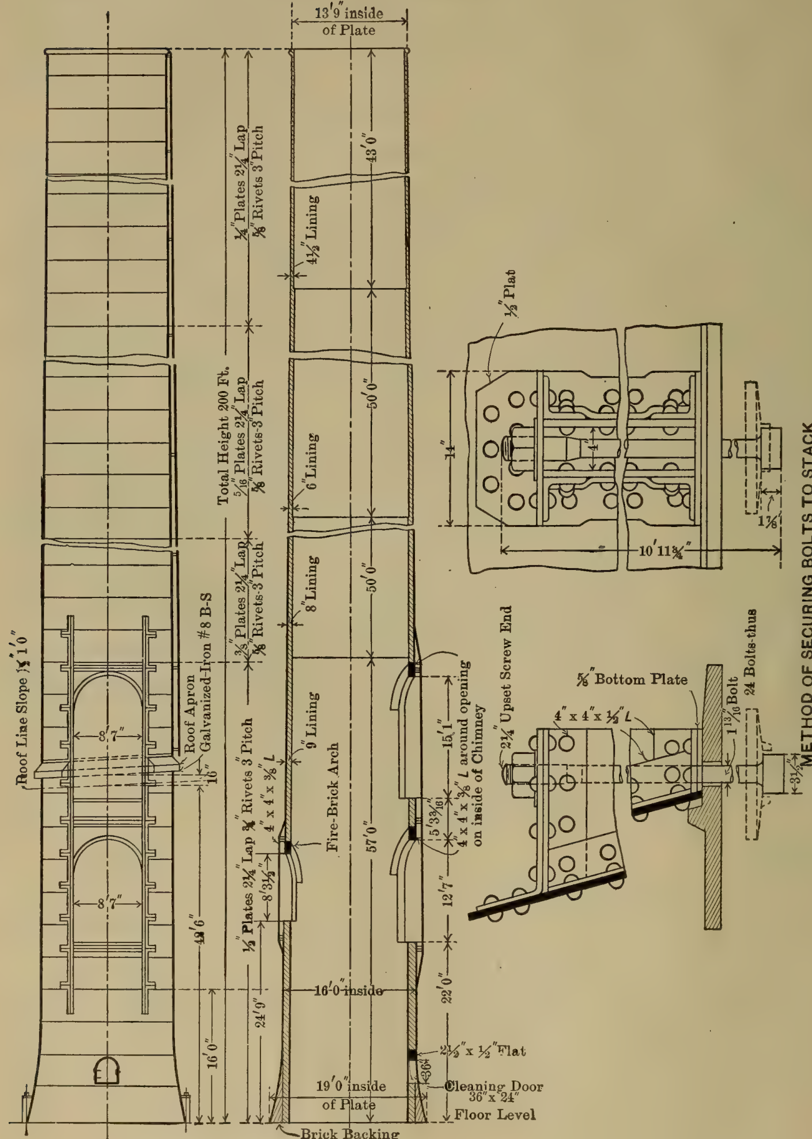


FIG. 158. Steel Chimney of the South Side Elevated Railway Power House, Chicago.

For chimneys under 7 feet in diameter and 150 feet in height the thickness of plate should not be less than $\frac{3}{16}$ inch, nor less than $\frac{1}{4}$ inch for larger sizes.

It is customary to make the courses about 5 feet in height for convenience in erection.

Table 47 gives the dimensions of self-supporting steel stacks as made by the Riter Conley Company of Pittsburg, who use the following empirical formula in determining the thickness of the shell,

$$S_1 = \frac{M}{0.8\,D_1^2}, \tag{66}$$

in which

S_1 = stress per lineal inch of section considered,
M = wind moment in inch-pounds, and
D_1 = diameter of the shaft in inches.

Allowing 8000 pounds per square inch as the safe stress for single-riveted joints and 10,000 for double-riveted joints, the required thickness is found by dividing S_1 by 8000 or 10,000.

Example: Determine the thickness of plate at a section 150 feet from the top of a cylindrical steel stack 12 feet in diameter and 200 feet high. Horizontal seams to be double riveted.

The total wind pressure on the section is

$$150 \times 12 \times 25 = 45{,}000 \text{ pounds.}$$

The moment arm is

$\frac{150}{2} \times 12 = 900$ inches.

$D_1 = 144$ inches; $S = 8000$ pounds per square inch.

TABLE 47.

STEEL STACKS. — SIZES OF RITER CONLEY COMPANY, PITTSBURG.

Diameter of Flue. Ft.	In.	Total Height. Ft.	Total Weight. Lb.	How Made.
5	6	165	67,000	40 ft. of $\frac{3}{16}$ in., 45 ft. of $\frac{1}{4}$ in., 50 ft. of $\frac{5}{16}$ in., 30 ft. of $\frac{3}{8}$ in.
7	0	160	79,000	30 ft. of $\frac{3}{16}$ in., 50 ft. of $\frac{1}{4}$ in., 50 ft. of $\frac{5}{16}$ in., 30 ft. of $\frac{3}{8}$ in.
8	6	150	94,000	60 ft. of $\frac{1}{4}$ in., 60 ft. of $\frac{5}{16}$ in., 30 ft. of $\frac{3}{8}$ in.
10	0	200	150,000	90 ft. of $\frac{1}{4}$ in., 60 ft. of $\frac{5}{16}$ in., 50 ft. of $\frac{3}{8}$ in.
12	0	200	175,000	35 ft. of $\frac{1}{4}$ in., 35 ft. of $\frac{9}{32}$ in., 35 ft. of $\frac{5}{16}$ in., 35 ft. of $\frac{11}{32}$ in., 35 ft. of $\frac{3}{8}$ in., 25 ft. of $\frac{13}{32}$ in.
11	6	225	232,000	40 ft. of $\frac{1}{4}$ in., 40 ft. of $\frac{9}{32}$ in., 40 ft. of $\frac{5}{16}$ in., 40 ft. of $\frac{11}{32}$ in., 40 ft. of $\frac{3}{8}$ in., 25 ft. of $\frac{7}{16}$ in.
12	0	255	256,000	75 ft. of $\frac{1}{4}$ in., 65 ft. of $\frac{5}{16}$ in., 55 ft. of $\frac{3}{8}$ in., 35 ft. of $\frac{7}{16}$ in., 25 ft. of $\frac{1}{2}$ in.

Substituting these values in equation (65),

$$Ph = S\frac{\pi}{32}\left(\frac{D_1^4 - D_2^4}{D_1}\right),$$

$$\frac{45{,}000 \times 1800}{2} = 8000 \times \frac{3.14}{32}\left(\frac{144^4 - D_2^4}{144}\right),$$

$$D_2 = 143.36.$$

Now
$$t = \frac{D_1 - D_2}{2}$$
$$= \frac{144 - 143.36}{2}$$
$$= 0.32 \text{ inch.}$$

The nearest commercial size lies between nine-thirty-seconds and five-sixteenths.

The Riter Conley formula gives for this section

$$S_1 = \frac{M}{0.8\,D_1^2} = \frac{45{,}000 \times 900}{0.8 \times 144^2}$$
$$= 2440 \text{ pounds,}$$
$$t = \frac{S_1}{8000} = \frac{2440}{8000} = 0.305 \text{ inch.}$$

151. Riveting. — The diameter of rivets should always be greater than the thickness of the plate but never less than one-half inch. The pitch should be approximately $2\frac{1}{2}$ times the diameter of the rivet, and always less than 16 times the thickness of the plate. Single-riveted joints are ordinarily used on all sections except the base, where the joint should be double riveted with rivets staggered, although in very large stacks all horizontal seams are double riveted to give greater stiffness to the shaft.

152. Stability of Steel Chimneys. — The wind being ordinarily the only force tending to overturn the stack, and the chimney being rigidly bolted to the foundation, a condition of stability requires that

$$(W_c + W_F)\frac{D}{3} \text{ be equal to or greater than } P\left(\frac{H}{2} + h\right), \qquad (67)$$

in which

W_c = weight of the chimney in pounds,
W_F = weight of the foundation,
P = total wind pressure in pounds,
D, H, and h, in feet, as indicated in the figure.

Expressed graphically: Lay off GP, Fig. 159, equal to the total wind pressure in direction and amount and acting at the center of pressure of the shaft; lay off GW equal to the weight of the stack and foundation; find the resultant GR and produce it to intersect the base line as at R'; if R' falls within the inner third of the base the stack is stable, provided, of course, that the chimney is properly designed and constructed. Therefore the heavier the combined weight of the chimney and its foundation the more stable the structure. (See also paragraph 157.)

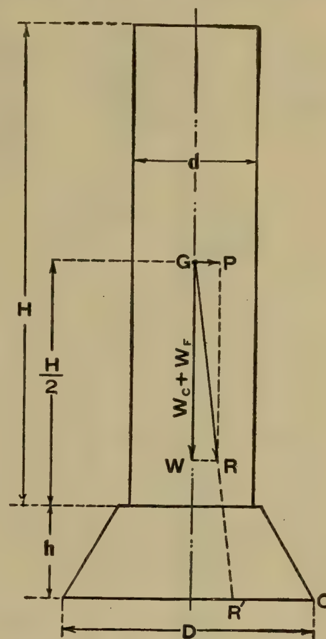


Fig. 159.

D in Fig. 159 varies from one-tenth to one-fifteenth H, depending upon the character of the subsoil. For the ordinary concrete foundation, Christie ("Chimney Design and Theory," p. 57) gives as an average value for D

$$D = \frac{H^2d}{26{,}000} + 10. \qquad (68)$$

Steel Chimneys: Elec. Rev., Apr. 7, 1911.

153. Brick Chimneys. — By far the greater number of power-plant chimneys are of brick construction and usually of circular section, though octagonal, hexagonal, and square sections are quite common. The round chimney requires the least weight for stability, and the others in the order mentioned. Taking the total wind pressure on the flat surface of a square stack as unity, the effective pressure, according to Rankine, for the same projected area will be 0.75 for the hexagonal, 0.6 for the octagonal, and 0.5 for the round. Henry Adams, Industrial Engineering, March, 1912, p. 199, states that these figures are not in accord with modern experiments, and gives the following multipliers: for a round chimney, 0.785; for an octagonal chimney, 0.82.

Brick chimneys may be divided into two general classes:

1. Single shell, Fig. 160, and
2. Double shell, Fig. 162.

The double shell is the more common and consists of an outer shaft of brickwork and an inner core or lining extending part way or throughout the entire length of the shaft.

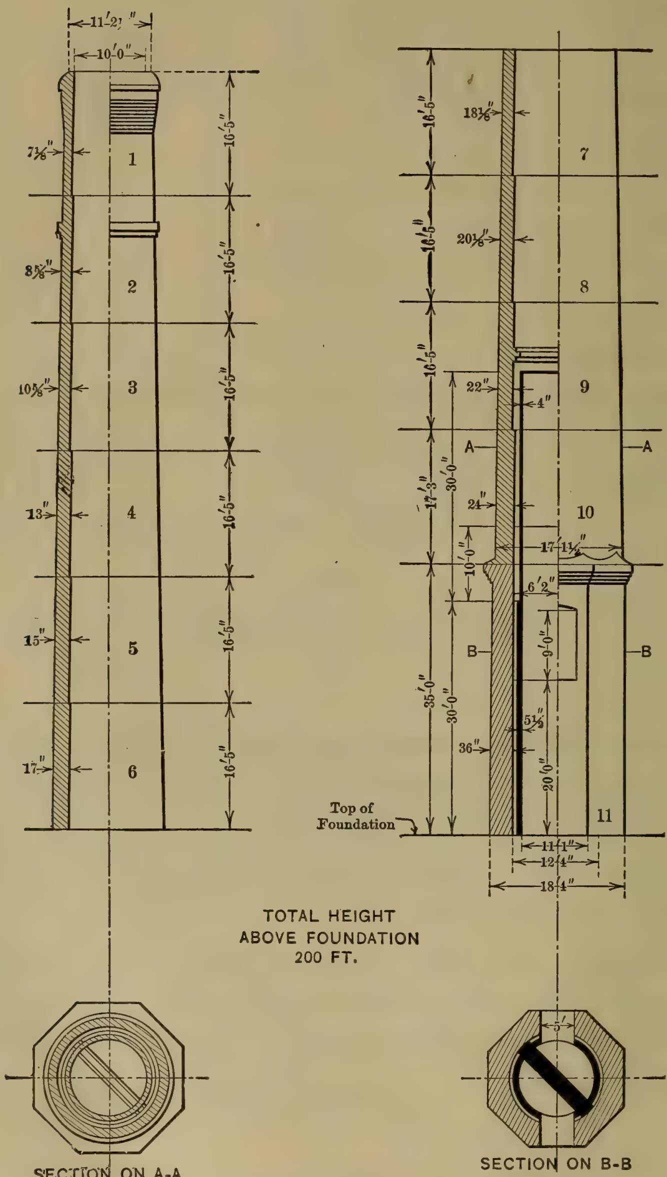


FIG. 160. Custodis Radial Brick Chimney.

The single shell is the general construction where carefully burned and selected brick not easily affected by the heat are used. As the inner core or lining is independent of the outer shell and has no part in the strength of the chimney, the rules for determining the thickness of the walls are practically the same for both single and double shell.

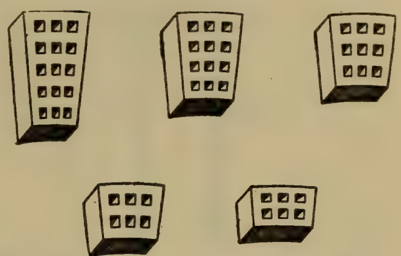


Fig. 161. Custodis Radial Brick.

154. Thickness of Walls. — The thickness of the wall should be such as to require minimum weight of material for the proper degree of stability, due consideration being paid to the practical requirements of construction. The thickness does not vary uniformly, but decreases from bottom to top by a series of steps or courses as in Fig. 163. In general, the thickness at any section should be such that the resultant stress of wind and weight of shaft will not put the masonry in tension on the windward side or in excessive compression on the leeward side.

For circular chimneys using common red brick for the outer shell the following approximate method gives results in conformity with average practice:

$$t = 4 + 0.05\,d + 0.0005\,H, \qquad (69)$$

where

t = thickness in inches of the upper course, neglecting ornamentation, and should, of course, be made equal to the nearest dimension of the brick in use. Ordinary red bricks measure $8\frac{1}{4} \times 4 \times 2$,

d = clear inside diameter at the top, inches.

H = height of stack, inches.

Beginning at the top with this thickness, add one-half brick, or 4 inches, for each 25 or 30 feet from the top downwards, using a batter of 1 in 30 to 1 in 36.

The minimum value of t for stacks built with inside scaffolding should be 7 inches for radial brick and $8\frac{1}{4}$ inches for common brick, as a thinner wall will not support the scaffold. Radial brick for chimneys are made in several sizes, so that the thickness of the walls when they are used increases by about 2 inches at the offsets.

For specially molded radial brick or for circular shells reënforced as in Fig. 162 the length of the different courses may be much less than stated above. The external form of the top is a matter of appearance, and may be designed to suit the taste, but should be protected by a

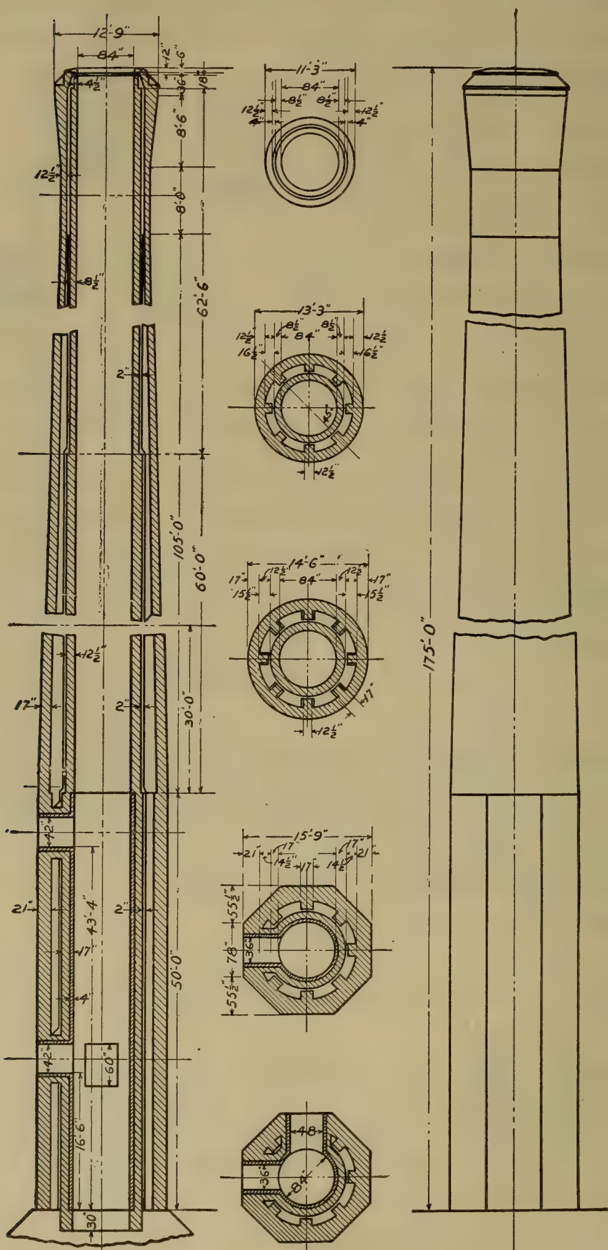


FIG. 162. Brick Chimney at the Power Plant of the Armour Institute of Technology.

cast-iron or tile cap and provided with lightning rods. Ladders for reaching the top of the chimney are generally located inside of brick stacks and outside of steel ones.

Professor Lang's rule (Eng. Rec., July 20, 1901, p. 53) for determining the length of the different courses is (Fig. 163)

$$h = C\left(20\,t + 60\,i + 0.1056\,G + 2.5\frac{d}{2} + 656\tan\alpha - 0.007\,H - 0.453\,p - 18.7\right), \tag{70}$$

in which

h = length of the course under consideration,
C = constant = 1 for a circular, 0.97 for an octagonal, and 0.83 for a square, chimney,
i = increase in thickness for each succeeding section in feet,
G = weight per cubic foot of brickwork,
p = wind pressure, pounds per square foot,
α = angle of the internal batter.
All other notations as indicated in Fig. 163.

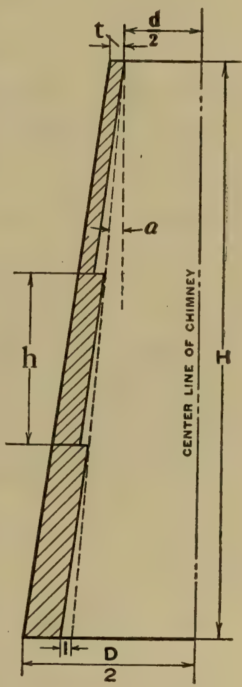


Fig. 163.

For chimneys over 100 feet in height he recommends that 100 be used instead of the actual height, since the critical point will be in one of the lower sections and not at the base.

If a value of h is obtained which is not contained an even number of times in H, it may be slightly increased or decreased so as to effect this result.

To determine the stresses at any section the shaft is treated as a cantilever uniformly loaded with a maximum wind pressure of 25 pounds per square foot. If the tension on the windward side subtracted from the compression leaves a positive remainder, the chimney will be stable; if the remainder is negative, the masonry will be in tension, which it withstands but feebly. The sum of the compressive stresses on the leeward side due to wind pressure and weight must be less than the crushing strength of the masonry. The practice, however, of assuming a fixed value for allowable pressure irrespective of the height of the stack gives dimensions that are too low for small stacks and too high for large stacks. According to Professor Lang, compressive stress on the leeward side in pounds per square inch with single chimneys should not exceed

$$p = 71 + 0.65\,L, \tag{71}$$

where

p = pressure in pounds per square inch,
L = distance in feet from top of chimney to the section in question.

With double shell $$p = 85 + 0.65\,L. \tag{72}$$

The tension on the windward side should not exceed,

$$\text{for single shell: } p = (18.5 + 0.056\,L), \tag{73}$$
$$\text{for double shell: } p = (21.3 + 0.056\,L). \tag{74}$$

Example: Determine the maximum stress in the outer fibers of the brickwork at the base of section 8 of the chimney illustrated in Fig. 160 when the wind is blowing 100 miles an hour.* Assume the weight of the brickwork 120 pounds per cubic foot.

A wind velocity of 100 miles per hour is estimated to exert a pressure of 50 pounds per square foot on a flat surface and approximately 25 pounds per square foot of projected area on a cylindrical surface. The height of the chimney to section 8 is 131.4 feet. The projected area as computed from the figure is 1800 square feet. Hence p, the total wind pressure, is 1800 × 25 = 45,000 pounds. The volume of brickwork above section 9 may be calculated, and is 6150 cubic feet, hence the weight W = 6150 × 120 = 738,000 pounds.

The area of the joint at this section is 75.3 square feet, therefore the pressure due to the weight of the superimposed brickwork is 738,000 divided by 75.3 = 9800 pounds per square foot. To find the stress due to the wind pressure, substitute the proper values in equation (65):

$$Ph = S\frac{I}{e} = 0.0983\left(\frac{D_1^4 - D^4}{D_1}\right)S.$$

Here

P = 45,000 as computed above,
h = 55 feet (found by laying out the section and locating the center of gravity),
D_1 = 16.2,
D = 12.9,

whence

$$45{,}000 \times 55 = 0.0983\,\frac{16.2^4 - 12.9^4}{16.2}\,S,$$

from which $$S = 9907 \text{ pounds per square foot.}$$

* A serious difference of opinion exists as to the effective pressure of wind on chimneys of different shapes, but in lieu of accurate experimental data to the contrary the figures given herewith may be used with confidence, since a vast number of stacks based upon the figures are successfully withstanding gales of from 60 to 80 miles an hour.

The net stress on any part of the section is the resultant of that due to the weight of the stack and that caused by the wind, the net stress on the windward side being

$$9907 - 9800 = 107 \text{ pounds per square foot,}$$

which is evidently a tensile stress and should never exceed the value given by formula (73):

$$\begin{aligned} p &= (18.5 + 0.056\,L) \\ &= (18.5 + 0.056 \times 131.4) \\ &= 25.8 \text{ pounds per square inch} \\ &= 3715 \text{ pounds per square foot.} \end{aligned}$$

The net compressive stress on the leeward side is $9800 + 9907 = 19{,}707$ pounds per square foot, which should not exceed that given by formula (71):

$$\begin{aligned} p &= 71 + 0.65\,L \\ &= 71 + 0.65 \times 131.4 \\ &= 156.4 \text{ pounds per square inch} \\ &= 22{,}521 \text{ pounds per square foot.} \end{aligned}$$

(See also analysis of steel-concrete chimney, paragraph 159.)

155. Core and Lining. — The core or lining of a brick chimney is commonly carried to the top of the shaft, though it sometimes extends only part of the distance. The inside diameter is generally uniform, the offsets being made on the outside. The core and outer shell should be independent to prevent injury due to expansion of the core. The rules for the thickness of lining in steel chimneys apply also to brick chimneys. The batters for the inner and outer shells should be such as to allow at least 2 inches clearance between the two shafts at the top, and the top should be protected by an iron ring or by a projecting ledge from the outer shell.

156. Materials for Brick Chimneys. — Brick for the external shaft should be hard burned, of high specific gravity, and laid with lime mortar strengthened with cement. Lime mortar itself is more resistant to heat, but hardens slowly and may cause distortion in newly erected stacks, and hence should be used only when a long time is taken in building. Mortar of cement and sand alone is not to be recommended, since it does not resist heat well and is attacked by carbon dioxide, particularly in the presence of moisture. A mortar consisting of 1 part by volume of cement, 2 of lime, and 6 of sand may be used for the upper brickwork, 1, $2\frac{1}{2}$, and 8 respectively for the lower part, and 1, 1, and 4 respectively for the cap. The harder the brick the more cement is necessary, as lime does not cling so well to

hard, smooth surfaces. The inner core may be constructed of second-class fire brick, since the temperature seldom exceeds 600 degrees F. Lime mortar is invariably used for the core.

157. Stability of Brick Chimneys. — When there is no wind blowing and the chimney is built symmetrically about a vertical axis the pressure due to weight is uniformly distributed over the bearing surfaces, and the center of pressure lies in the line *XX*, Fig. 164. But when the wind blows the pressure exerted tends to tilt the shaft as a whole column in the direction of the current, and the resultant pressure at the windward side of the base decreases, until, with a sufficiently high velocity of wind, it may become zero, in which case the center of pressure moves a distance q towards the leeward side of the base. As soon as the pressure at A becomes zero the joint begins to open (assuming no adhesion between chimney and base) and the shaft is evidently in the condition of least stability. The distance q through which the center of pressure has moved is called the *radius of the statical moment.* For any column it may be shown that

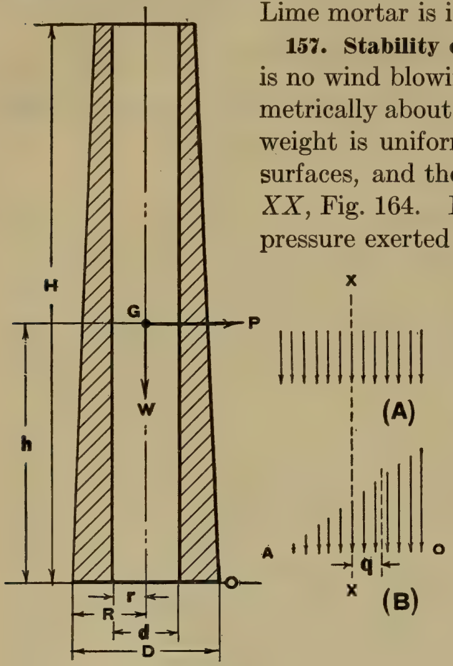


Fig. 164.

$$q = \frac{I}{Ae} \text{ (Rankine, "Applied Mechanics," p. 229),} \qquad (75)$$

in which

I = moment of inertia of the section,
A = area of the section,
e = distance from the center of the shaft to the outer edge of the joint.

Thus for circular section, $q = \dfrac{D}{8}$.

For square section, $q = \dfrac{D}{6}$.

For annular circular ring, $q = \dfrac{D^2 + d^2}{8D}$.

For hollow square, $q = \dfrac{D^2 + d^2}{6D}$.

The relationship between weight of shaft and wind pressure for the condition of least stability is

$$Ph = Wq, \tag{76}$$

in which

P = total wind pressure, pounds,

h = distance in feet from the base line of the section under consideration to center of gravity of that section,

W = weight of shaft in pounds above the assumed base line,

q = radius of the statical moment

The condition of least stability for round chimneys requires, therefore, that

$$Ph = W\frac{D^2 + d^2}{8D}. \tag{77}$$

For many purposes it is sufficiently accurate to assume $D = d$, and equation (77) becomes

$$Ph = W\frac{D}{4} \text{ for round chimneys,} \tag{78}$$

$$Ph = W\frac{D}{3} \text{ for square chimneys.} \tag{79}$$

The rule commonly used in Germany, and which is finding some favor with engineers in the United States, gives for the condition of least stability

$$W\left(\tfrac{1}{2}R + \tfrac{1}{4}r\right) = Ph. \quad \text{(Eng. Rec., July 27, 1901, p. 82.)} \tag{80}$$

Notations as in Fig. 162, all dimensions in feet.

This permits of a lighter chimney than equation (77), and the maximum wind pressure may be assumed to put the joint on the windward side in tension or even to permit a slight opening of same.

A rule of thumb for stability is to make the diameter of the base one-tenth of the height for a round chimney; for any other shape to make the diameter of the inscribed circle of the base one-tenth of the height.

The factor of stability is the quotient obtained by dividing the value of q from formula (76) by that from (75). If less than unity, the chimney is in tension at the outer fiber on the windward side, and must be redesigned unless the tension is less than that allowed by equation (73). Calculations for stability should be made for various sections.

Example: Analyze the chimney illustrated in Fig. 160 for stability at, say, section 8, the following data referring to the portion above the base line of this section.

From the drawing:
Projected area of the stack, 1800 square feet.
Volume of brickwork, 6150 cubic feet.
Outside diameter of base, 16.2 feet.
Inside diameter of base, 12.9 feet.
Center of pressure to base line, 55 feet.
Total height above base line, 131.4 feet.
Maximum total wind pressure:

$$P = 1800 \times 25 \doteq 45{,}000 \text{ pounds.}$$

Weight of shaft:

$$W = 6150 \times 120 = 738{,}000 \text{ pounds.}$$

For stability, according to equation (55),

$$Ph < W \frac{D^2 + d^2}{8\,D}.$$

Substituting the proper values:

$$Ph = 45{,}000 \times 55 = 2{,}475{,}000 \text{ foot-pounds.}$$

$$W \frac{D^2 + d^2}{8\,D} = 738{,}000 \left(\frac{\overline{16.2^2} + \overline{12.9^2}}{8 \times 16.2}\right) = 2{,}441{,}000.$$

While Ph is slightly greater than $W \dfrac{D^2 + d^2}{8\,D}$, for practical purposes the shaft at this section would be called stable under maximum allowable wind pressure.

For stability, according to equation (80),

$$Ph < W\left(\tfrac{1}{2} R + \tfrac{1}{4} r\right),$$

$$Ph = 2{,}475{,}000, \text{ as determined above,}$$

$$W\left(\frac{1}{2} R + \frac{1}{4} r\right) = 738{,}000\left(\frac{8.1}{2} + \frac{6.45}{4}\right)$$

$$= 4{,}177{,}000.$$

Ph is therefore considerably less than $W\left(\tfrac{1}{2} R + \tfrac{1}{4} r\right)$, and the condition imposed in equation (80) is more than fulfilled.

The Design of Tall Chimneys: Henry Adams, Industrial Engineering, March, 1912, p. 198. *Design of a Brick Chimney:* Eng. News, May 9, 1912, p. 866.

158. Custodis Radial Brick Chimney. — Fig. 160 gives the details of a 200 × 10-foot radial brick chimney constructed of special molded radial brick, formed to suit the circular and radial lines of each section, thus permitting them to be laid with thin, even mortar joints. The blocks are much larger than common brick and the number of joints is proportionately reduced. They are molded with vertical perforations, as shown in Fig. 161, which permits thorough burning, thereby increasing the density and strength and at the same time reducing the weight of the block. In laying, the mortar is worked into the perforations about one-half inch. The first 60 feet above the base are octagonal in section, with 36-inch walls, and the balance of circular section, with walls tapering gradually from 22 inches to $7\frac{1}{8}$ inches in thickness. A radial brick lining extends 60 feet from the base as indicated. The chimney was designed to furnish draft for a 3500-horsepower boiler plant and cost, erected, $8,800. The entire weight of the chimney exclusive of foundation is 870 tons.

Radial brick chimneys without the inner lining are likely to be unduly affected by heat.

The tallest chimney in the world (1912), located at Great Falls, Mont., is of the Custodis type, and is used for leading off the gases from the smelter plant of the Boston and Montana Consolidated Copper and Silver Mining Company. The height above the top of the foundation is 506 feet, and the internal diameter at the top 50 feet. The chimney and foundation cost approximately $200,000.

Custodis Chimney Details: Eng. Rec., Oct. 1, 1904, p. 385; Power, May, 1900, p. 12.

159. Steel-Concrete Chimneys. — The use of concrete reënforced with iron or steel for the construction of chimneys is rapidly increasing. The advantages claimed for this class of stack are:

1. Light weight of the whole structure, being but one-third as great as an equivalent common brick chimney. The space occupied is much less than with either brick or steel stack, on account of the thinness of walls at the base and the absence of any flare or bell.
2. Total absence of joints, the entire structure including foundation being a monolith.
3. Great resisting power against tension and compression.
4. Rapidity of construction. May be erected at an average rate of six feet per day.
5. Adaptability of the material to any form.

This type of chimney being comparatively new, little data concerning depreciation are available, but some which have been in use ten years show little or no deterioration.

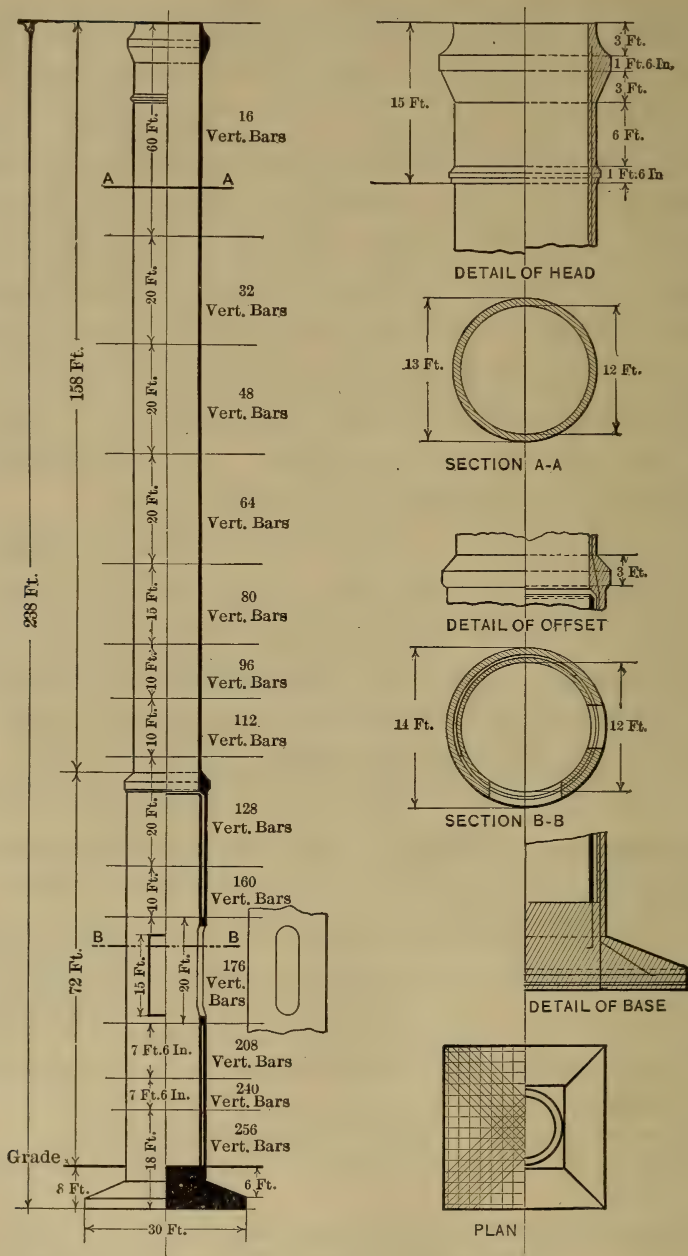


FIG. 165. Weber Reënforced Concrete Chimney.

Fig. 165 gives the details of a Weber steel-concrete chimney erected at Portland, Ore., for the Portland General Electric Company. The entire structure, foundation and shaft, is a monolith, 238 feet in total height and 12 feet internal diameter, weighing only 889 tons. It occupies but 168 square feet of ground space at the grade level. The weight not including foundation is 470 tons. The stack was erected complete in 58 working days, and cost approximately $13,000.

The cement used was German Portland mixed with select bank sand in proportion of one to three, gravel or crushed stone being used only in the foundation below the ground. The mortar was used medium dry and tamped in the form around the steel reënforcement.

The shaft is of the double-shell type, with inner core extending 70 feet above the grade. The core is but 4 inches in thickness at the base, and the outer shell 8 inches. Both inner and outer shell are reënforced with vertical T bars, $1\frac{1}{4} \times 1\frac{1}{4} \times \frac{3}{16}$ inch, of low-carbon Bessemer steel, spaced at the base 24 inches between centers in the inner core and 4 inches in the outer shell, and increasing in spacing to the top, where the distance between the bars is 12 inches. The horizontal rings are $1 \times 1 \times \frac{1}{8}$ T's spaced 18 inches between centers in the core and 36 inches in the outer shell. The steel bars vary from 16 to 30 feet in length, and where they meet lengthwise are lapped not less than 24 inches. The use of different lengths of steel prevents the laps from concentrating in any given section.

The tallest chimney of this type (1907) was erected for the Butte Reduction Works at Butte, Mont. Its height is 350 feet and inside diameter 18 feet.

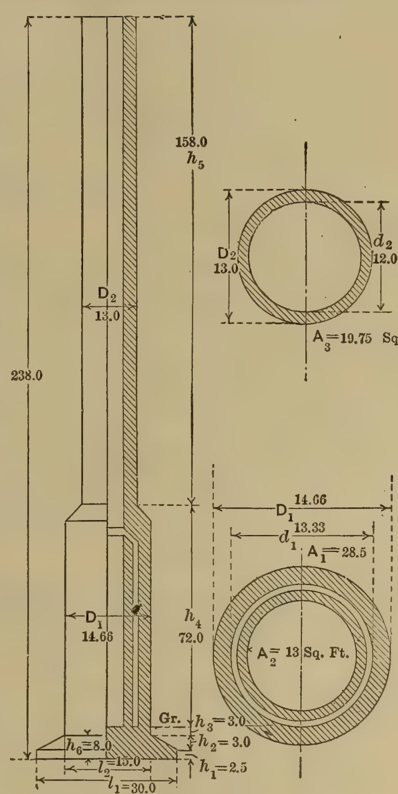


Fig. 166.

The following strain sheet gives the Weber Company's analysis of the chimney illustrated in Fig. 165, and is based on a wind pressure of 50 pounds per square foot. Notations as in Fig. 166.

WEIGHTS.

W_f = weight of foundation

$$= \left(l_1^2 h_1 + \frac{l_1^2 + l_2^2}{2} h_2\right) 150$$

$$= \left(30^2.2 + \frac{30^2 + 15^2}{2} 3\right) 150$$

= 523,200 pounds.

150 = weight per cubic foot of concrete.

W_e = earth weight on foundation

$= \{l_1^2 h_6 -$ (volume of foundation)$\}$ 100

= (7200 cubic feet − 3995 cubic feet) 100

= 320,500 pounds.

100 = weight per cubic foot of earth.

W = weight of shaft

$$= \{A_1 (h_4 + h_3) + A_2 (h_4 + h_3) + A_3 h_5\} 150$$

$$= \{38.5 (72 + 3) + 13 (72 + 3) + 19.75 \cdot 158\} 150$$

= 934,950 pounds.

W_t = total weight

$= W_f + W_e + W$ = 1,778,650 pounds (889 tons).

SECTION AT GRADE G_r.

I. W_i = weight of outer shell and single shell above section

$$= (A_1 h_4 + A_3 h_5) 150$$

$$= (28.5 \cdot 72 + 19.75 \cdot 158) 150$$

= 775,806 pounds.

II. r = radius of statical moment

$$= \frac{D_1}{8}\left[1 + \left(\frac{d_1}{D_1}\right)^2\right]$$

$$= \frac{14.66}{8}\left[1 + \left(\frac{13.33}{14.66}\right)^2\right]$$

= 3.35 feet.

III. P = wind pressure on chimney

$$= D_1 h_4 \frac{50}{2} + D_2 h_5 \frac{50}{2}$$

$$= 14.66 \times 72 \times 25 + 13 \times 158 \times 25$$

= 77,738 pounds.

M = wind moment on section

$$= D_1 h_4 \frac{50}{2} \times \frac{h_4}{2} + \left(D_2 h_5 \frac{50}{2}\right)\left(h_4 + \frac{h_5}{2}\right)$$

$$= 14.66 \times 72 \times \frac{50}{2} \times \frac{72}{2} + \left(13 \times 158 \times \frac{50}{2}\right) 72 + \frac{158}{2}\Big)$$

= 8,703,818 foot-pounds.

IV. N = statical moment
$= rW_i$
$= 3.35 \times 775{,}806$
$= 2{,}598{,}950$ foot-pounds.

V. B = bending moment
$= M - N$
$= 8{,}703{,}818 - 2{,}598{,}950$
$= 6{,}104{,}868$ foot-pounds.

VI. $\frac{I}{e}$ = section modulus

$$= 0.0982\left(\frac{D_1^4 - d_1^4}{D_1}\right)$$

$$= 0.0982\left(\frac{\overline{14.66 \times 12}^4 - \overline{13.33 \times 12}^4}{14.66}\right)$$

$= 169{,}703.$

VII. z = tension per square inch sectional area

$$= 12\,B \div \frac{I}{e}$$

$= 12 \times 6{,}104{,}868 \div 169{,}703$
$= 432.5$ pounds.

VIII. Z = total tension
$= 144\,A_1 z$
$= 144 \times 28.5 \times 432.5$
$= 1{,}825{,}015$ pounds.

IX. s = area steel required

$= \frac{Z}{a}$. (a = sectional strain on steel)
= 16,000 pounds per square inch
$= 114.2$ square inches.

X. K = number of bars

$= \frac{s}{x}$. (x = 0.45 square inch = area of one bar)
$= 252$ bars.

FOR STABILITY.

XI. L = length of one side of base.

$$L = \frac{M}{W_t} \times 6$$

$$= \frac{8{,}703{,}818}{1{,}778{,}650} \times 6$$

$= 29.4$ feet.

SECTION 42′ 0″ ABOVE GRADE.

I. W_1 = 596,250 pounds.
II. r = 3.35 feet.
III. P = 62,295 pounds. M = 5,761,325 foot-pounds.
IV. N = 1,997,438 foot-pounds.
V. B = 3,763,889 foot-pounds.
VI. $\frac{I}{e}$ = 169,703.
VII. z = 222 pounds.
VIII. Z = 911,088 pounds.
IX. s = 57 square inches.
X. K = 127 bars.

SECTION AT OFFSET.

I. W_1 = 468,000 pounds.
II. r = 3 feet.
III. P = 51,350 pounds. M = 4,056,650 foot-pounds.
IV. N = 1,404,000 foot-pounds.
V. B = 2,652,650 foot-pounds.
VI. $\frac{I}{e}$ = 102,041.
VII. z = 311 pounds.
VIII. Z = 786,000 pounds.
IX. s = 55.5 square inches.
X. K = 123 bars.

SECTION 50′ 0″ FROM TOP.

I. W_1 = 148,125 pounds.
II. r = 3 feet.
III. P = 16,250 pounds. M = 406,250 foot-pounds.
IV. N = 444,365 foot-pounds.

Since the statical moment N is greater than the wind moment M, there is no bending moment B, so no steel is required, the chimney above this section standing of its own weight. However, thirty-two bars are continued to the top.

Analysis of Reënforced Concrete Chimneys: Eng. Rec., Apr. 8, 1911; Jan. 13, 1912.

160. Breeching. — The area of the flue or breeching leading from the boilers to the chimney is generally made equal to or a little larger than the internal area of the chimney, 20 per cent greater being an average figure. The flue may be carried over the boilers or back of the setting or even under the fire-room floor, but in any case should be as short as

possible and free from abrupt turns. Short right-angled turns reduce the draft approximately 0.05 inch for each turn, and a convenient rule is to allow 0.1 inch loss for each 100 feet of flue if of circular cross section and constructed of steel, and double this amount for brick flues of square section. The cross section of the flue need not be the same throughout its entire length, but may be tapered and proportioned to the number of boilers. Where two flues enter the stack on opposite sides, a diaphragm is inserted as indicated in Fig. 160. Flues should be covered with heat-insulating material.

161. Chimney Foundations. — On account of the concentration of weight on a small area the foundation of a chimney should be carefully designed. In most cities the building laws limit the maximum loads allowed for various soils and materials, and although they vary considerably the average is approximately as follows:

Material.	Safe Load, Lb. per Sq. Ft.
Hard-burned brick masonry, cement mortar, 1 to 2	20,000–30,000
Hard-burned brick masonry, cement mortar, 1 to 4	18,000–24,000
Hard-burned brick masonry, lime mortar	10,000–16,000
Concrete, 1 to 8	8,000–10,000

Kind of Soil.	Safe Load, Tons per Sq. Ft.
Quicksands and marshy soils	0.5
Soft wet clay	1.0
Clay and sand 15 feet or more in thickness	1.5
Pure clay 15 feet or more in thickness	2.0
Pure dry sand 15 feet or more in thickness	2.0
Firm dry loam or clay	3.0– 4.0
Gravel well packed and confined	6.0– 8.0
Rock broken but well compacted	10.0–15.0
Solid bed rock	Up to ½ of its ultimate crushing strength.

	Tons per Pile.
Piles in made ground	2.0
Piles driven to rock or hardpan	25.0

Chimney foundations as a rule are constructed of concrete except where the low sustaining nature of the soil necessitates the use of piles or a grillage of timber or steel. For masonry chimneys the foundation is designed to give the necessary support to the shaft without particular reference to its mass or distribution, as the shape of the foundation has virtually no effect on its stability as a column. In steel and reënforced concrete chimneys the shape and weight of the foundation are a function of the desired factor of stability, since the shaft is securely anchored to the foundation and the two form practically one mass. The foundation should be designed to fulfill the conditions in formula (68) in addition to the requirements for mere support.

Table 48 gives the least diameter and depth of foundation for steel chimneys of various diameters and heights.

162. Chimney Efficiencies. — The chimney as a *mover of air* has a very low thermodynamic efficiency. Compared with that of a fan its performance is very poor, and mechanical-draft concerns sometimes use this as an argument.

Example: A chimney 200 feet high and 10 feet in diameter furnishes draft for a battery of boilers rated at 3500 horse power. Average outside temperature 60 degrees F.; temperature of flue gases 500 degrees F.; calorific value of the fuel 14,000 B.t.u. per pound. Compare the thermal efficiency of the chimney as a mover of air with that of a forced-draft apparatus of equivalent capacity.

TABLE 48.

SIZES OF FOUNDATION FOR STEEL CHIMNEYS.

Diameter, Feet.	Height, Feet.	Least Diameter of Foundation.	Least Depth of Foundation.
3	100	15′ 9″	6′ 0″
4	100	16′ 4″	6′ 0″
4	125	18′ 5″	7′ 0″
5	150	20′ 4″	9′ 0″
5	200	23′ 8″	10′ 0″
6	150	21′ 10″	8′ 0″
6	200	25′ 0″	10′ 0″
7	150	22′ 7″	9′ 0″
7	250	29′ 8″	12′ 0″
9	150	23′ 8″	10′ 0″
9	275	33′ 6″	12′ 0″
11	250	24′ 8″	10′ 0″
11	350	36′ 0″	14′ 0″

From Table 42 we find that a chimney 200 feet high, with temperatures as stated above, will furnish a theoretical draft of 1.27 inches, equivalent to a pressure of 6.6 pounds per square foot. Neglecting friction the height H of a column of external air which would produce this pressure is

$$H = \left(\frac{d_1 - d}{d_1}\right) h, \tag{81}$$

in which

h = height of the chimney in feet,

d = density of the hot gases in the stack,

d_1 = density of the outside air.

Substitute in (59)

$$d_1 = 0.0763, \quad d = 0.0435, \quad \text{and} \quad h = 200.$$

$$H = \left(\frac{0.0763 - 0.0435}{0.0763}\right) 200$$

$$= 85.9 \text{ feet.}$$

The theoretical velocity of the air entering the base of the chimney under this head is

$$\begin{aligned} v &= \sqrt{2\,gH} \\ &= \sqrt{2 \times 23.2 \times 85.9} \\ &= 74.5 \text{ feet per second.} \end{aligned}$$

The weight of the gas escaping per second

$$\begin{aligned} &= 74.5 \times \text{area of the stack} \times 0.0763 \\ &= 446 \text{ pounds.} \end{aligned}$$

The displacement of this volume of gas is the result of heating it from 60 to 500 degrees F. Taking the specific heat of the gas as 0.2375, the heat necessary to displace 446 pounds per second is

$$\begin{aligned} \text{Heat required} &= 446 \times 0.2375 \times (500 - 60) \\ &= 46{,}500 \text{ B.t.u. per second.} \end{aligned}$$

The work actually performed is that of overcoming a total resistance of $6.6 \times 78.5 = 518$ pounds ($78.5 =$ internal area of the chimney) through a space of 74.5 feet; i.e.,

$$\begin{aligned} \text{Work done} &= 74.5 \times 518 = 38{,}591 \text{ foot-pounds per second} \\ &= 49.7 \text{ B.t.u. per second.} \end{aligned}$$

$$\text{Efficiency} = \frac{49.7}{46{,}500} = 0.00107, \text{ or about } \tfrac{1}{10} \text{ of 1 per cent.}$$

If a fan be substituted for the chimney and we allow say 8 per cent for the efficiency of engine and boiler, 40 per cent for the fan, and 25 per cent for friction, the combined efficiency will be

$$0.08 \times 0.40 \times 0.75 = 0.024, \text{ or 2.4 per cent.}$$

The fan then will be $\frac{0.024}{0.00107} = 22.4$ times more efficient than the chimney *as a mover of air.*

163. Cost of Chimneys. — Christie ("Chimney Design and Theory") gives the following costs of chimneys 150 feet high and 8 feet internal diameter:

Common red brick	approximate cost		$8,500.00
Radial brick	do	do	6,800.00
Steel, self-supporting, full lined	do	do	8,300.00
Steel, self-supporting, half lined	do	do	7,800.00
Steel, self-supporting, unlined	do	do	5,820.00
Steel, guyed	do	do	4,000.00

The following approximate costs of various sizes of a well-known radial brick chimney give an idea of the variation in cost due to increase in diameter and height:

Size of Chimney.		Cost.	Size of Chimney.		Cost.
Height.	Diameter.		Height.	Diameter.	
Feet.	Feet.		Feet.	Feet.	
75	4	$1,350.00	175	8	$7,050.00
75	6	1,950.00	175	10	7,925.00
75	8	2,650.00	175	12	8,950.00
75	10	3,725.00	175	14	9,725.00
125	6	3,500.00	200	8	9,250.00
125	8	4,250.00	200	10	10,500.00
125	10	4,675.00	200	12	11,100.00
125	12	5,125.00	200	14	12,500.00
150	8	6,150.00	250	10	16,500.00
150	10	7,125.00	250	12	18,250.00
150	12	7,750.00	250	14	21,500.00
150	14	8,275.00	250	16	24,250.00

TABLE 49.

PROPORTIONS OF CHIMNEYS FOR FACTORY STEAM BOILERS, COLLECTED FROM PRACTICE. (Hutton.)

Height of Chimney above the Ground in Feet.	Internal Dimensions.		Ratio of Bottom to Top. Internal Area.	Thickness of Walls.	
	Size of Base at the Ground Line.	Size of Top.		Thickness at Base in Inches at Ground Line.	Thickness at the Top in Inches.
40	2′ 6″	1′ 9″ sq.	2.04	18	9
60	2′ 11″	2′ 0″ sq.	2.12	18	9
70	3′ 4″	2′ 3″ sq.	2.13	23	9
80	3′ 8″	2′ 6″ sq.	2.18	28	9
90	4′ 0″	2′ 9″ sq.	2.27	28	9
100	4′ 8″	3′ 0″ diam.	2.40	28	9
110	4′ 10″	3′ 3″ diam.	2.33	28	9
120	5′ 6″	3′ 6″ diam.	2.40	28	9
135	6′ 0″	4′ 0″ diam.	2.30	28	9
150	4′ 6″	3′ 0″ diam.	2.25	28	14
155	6′ 0″	4′ 6″ diam.	1.78	56	14
160	9′ 0″	5′ 0″ sq.	3.24	36	14
170	7′ 6″	5′ 0″ diam.	2.25	36	14
180	6′ 4″	4′ 6″ diam.	2.00	54	14
200	5′ 3″	3′ 6″ diam.	2.28	36	14
225	16′ 0″	6′ 6″ sq.	4.00	36	14
250	19′ 0″	13′ 0″ diam.	2.13	40	14
300	14′ 0″	9′ 0″ diam.	2.42	48	14
450	21′ 6″	10′ 2″ diam.	4.35	59	14

CHAPTER VIII.

MECHANICAL DRAFT.

164. General. — The intensity of natural draft in a chimney depends mainly upon the height of the stack and the temperature of the chimney gases, and the chimney should be designed to meet the maximum requirements, permitting the damper to be partly shut at times. There is usually no practicable means of increasing natural draft *per se* after the maximum has been reached. Again, chimney draft is peculiarly susceptible to atmospheric influence and may be seriously impaired by adverse winds and air currents. Notwithstanding these apparent limitations, by far the greater number of steam power plants depend upon chimneys for draft because of the disposition of the waste gases. In many cases artificial draft has a great advantage and under certain conditions is indispensable; it is very flexible and readily adjusted to effect various rates of combustion, irrespective of climatic influences, and permits any degree of overload without undue expenditure of energy.

Artificial draft may be broadly classified under two heads:

1. The vacuum or induced draft; and
2. The plenum or forced-draft method.

In the former a partial vacuum is produced above the fire by suitable apparatus, and the effect is substantially that of natural draft.

In the forced-draft system pressure is produced in the ash pit, the air being forced through the grate.

In both systems the artificial draft is usually produced by either:

1. Steam jets; or
2. Centrifugal fans or exhausters.

165. Steam Jets. — Fig. 167 shows an application of a ring jet to the base of a stack. The apparatus is very simple, inexpensive, and easily applied. It consists essentially of a ring or a series of concentric rings of 1-inch or $1\frac{1}{2}$-inch pipe, perforated on the upper side with $\frac{1}{16}$, or $\frac{1}{8}$-inch holes, and placed in the base of the stack, so that the jets are discharged upward, thus creating a draft independent of the temperature of the flue gases. The steam connection to the jet is generally made direct to the boiler and not to the steam main, though the jet is often produced by exhaust steam.

Fig. 168 illustrates a Bloomsburg jet, which involves to some extent the principle of the ejector.

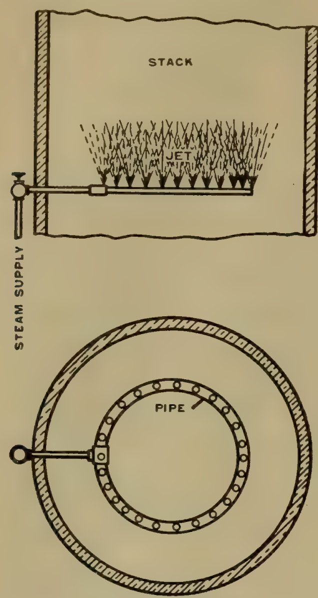


FIG. 167. Ring Steam Jet.

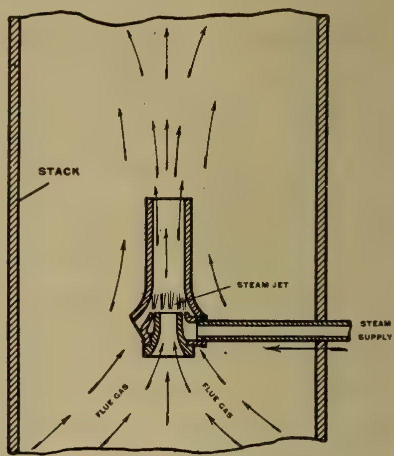


FIG. 168. Bloomsburg Jet.

The increase in draft produced by these devices as ordinarily installed is not great, although in locomotive practice where the entire exhaust is discharged up the stack an intense draft is obtained.

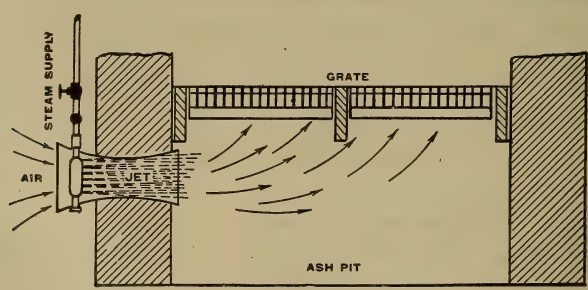


FIG. 169. McClaves Argand Blower.

Fig. 169 shows the application of a "McClaves argand blower." The steam is discharged below the grate through a perforated hollow ring, as indicated, drawing the air through the funnel by inspiration. This creates a powerful draft by forming an air pressure in the ash pit, and is an especially useful system of forcing fires for boilers which need forcing for short periods only.

Steam jets, as ordinarily installed in small plants, are very uneconomical, since a large amount of steam is required to produce good results. Table 50, based on experiments at the New York Navy Yard, to determine the best form of steam jet for producing draft in launch boilers,

TABLE 50.

RESULTS OF EXPERIMENTS UPON STEAM JETS AT NEW YORK NAVY YARD.*

Index of Jet.	Pounds of Water Evaporated per Hour.				
	A	B	C	D	E
In boiler making steam........	463.8	580.0	361.25	528.5	545.00
In boiler supplying jets.......	97.5	120	30	63.2	76.25
Per cent of steam used by jet......................	21.2	20.7	8.3	12.0	19.0

* Annual Report of the Chief of the Bureau of Steam Engineering, U. S. Navy, 1890.

shows steam consumptions of from 8.3 to 21.2 per cent of the total steam made. Table 51 gives the steam consumption of a number of types of steam jet blowers as determined by A. J. Whitman. The best performance is 4.6 per cent and the poorest 11.1 per cent of the total

TABLE 51.

CONSUMPTION OF STEAM BLASTS COMPARED.†

Coal.	Name of Blower.	Per Cent of Air Openings in Grate.	Pounds of Dry Coal burned per Hour per Square Foot of Grate.	Per Cent of Total Steam Generated in the Boilers that is required to operate the Steam Blasts.
Rice...........	Young.........	11	25.8	11.1
Do.........	....do..........	11	17.9	7.0
Do.........	Wilkinson.....	7	27.0	10.8
Buckwheat.....	Young.........	11	27.3	10.8
Do.........	....do..........	11	16.7	4.6
Do.........	....do..........	26	31.4	8.9
Do.........	McClave.......	11	16.4	6.7
Do	....do	11	26.1	9.3
Do.........	Wilkinson.....	7	32.5	7.8
Do.........	....do..........	7	45.4	10.2

† Trans. A.S.M.E., Vol. XVII. — See Whitham.

boiler steam generated. Steam jets below the grate are said to prevent clinkers from forming where fine anthracite coals are used, and thus to assist in keeping the fire free and open. They also assist in the economical combustion of certain low-grade fuels. See paragraph 102 for the influence of steam jets in effecting smokeless combustion.

The curves in Fig. 170 are of interest in showing the intensity of draft created by steam jets in the modern locomotive and the influence of the draft on the capacity of the boiler and the air supply. These curves are taken from Bulletin 21, U. S. Bureau of Mines, 1911.

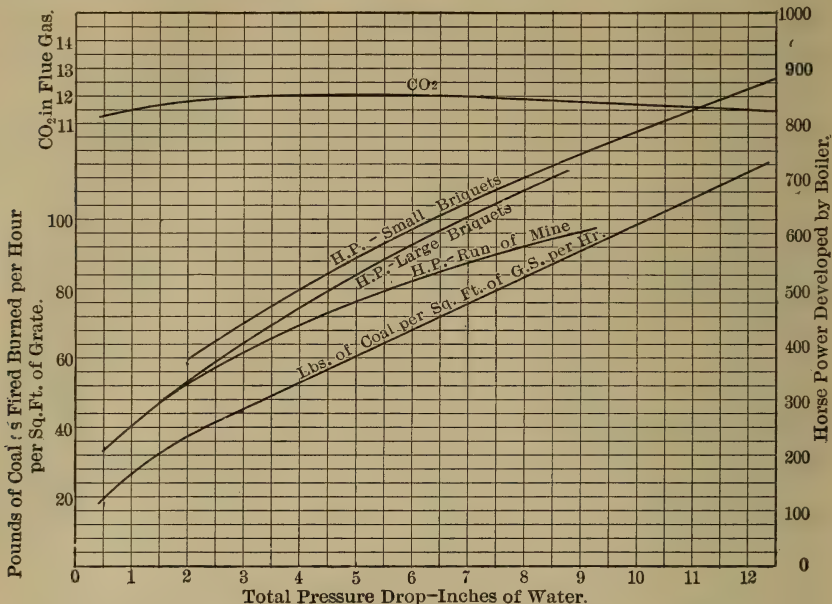


FIG. 170. Influence of Draft on the Performance of a Locomotive Boiler.

In large modern central stations where boiler overloads of from 150 to 250 per cent above rating are desirable, steam jets and mechanical blowing and stoking appliances use but a nominal percentage of the steam generated. The results in Table 52, taken from the tests of the

TABLE 52.

STEAM CONSUMPTION OF DRAFT APPLIANCES AND STOKER ENGINES, 2365 H.P. STIRLING BOILER.

(Delray Station, Detroit Edison Co.)

RONEY STOKER.

No. of Test.	Per Cent of Rating.	Dry Coal per Sq. Ft. G. S. per Hr.	Steam Consumption, Per Cent of Total Generated.			Draft, Inches of Water.		
			Stoker Engines.	Steam Jets.	Total.	Below Dampers.	In Furnace.	Ash Pit.
5	94	14.81	0.19	1.56	1.75	0.16	0.24	0.10
4	152	25.97	0.15	1.43	1.58	0.55	0.22	0.02
18	195.7	33.60	0.13	1.19	1.32	1.11	0.33	0.05

TABLE 52 — *Concluded.*

TAYLOR STOKER.

No. of Test.	Per Cent of Rating.	Dry Coal per Sq. Ft. G. S. per Hr.	Steam Consumption of Stoker Engines and Turbine Blower.	Draft, Inches of Water.		
				At Blast in Tuyeres.	Suction Below Boiler Dampers.	Suction in Ash Pit.
10	92.9	16.43	2.63	0.67	0.20	0.15
9	162.8	29.23	2.87	1.73	0.53	0.06
11	211.0	38.75	3.41	2.53	0.84	0.02

All of the steam exhausted from the Taylor equipment may be returned to the feed-water heater, whereas only that exhausted from the engines in the Roney equipment may be used in this manner, hence the *net heat* used is approximately the same in both cases.

For application of steam jets to mechanical stokers see Chapter IV.

large Stirling boilers at the Delray Station of the Detroit Edison Company, show what may be expected from installations of this class (Jour. A.S.M.E., Nov., 1911).

166. Fan Draft. — Fig. 171 shows a typical installation of a centrifugal fan on the forced-draft or plenum principle, the fan creating a pressure in the ash pit and forcing air through the fuel. The most approved method is to pass the air through the bridge wall, thence toward the front of the grate, though it may enter through an underground duct or through the side of the setting. Forced draft is usually adopted in old plants where increased demands for power require that the boilers be forced far above their rating to save the heavy expense of new boilers, or in plants burning refuse, anthracite culm or screenings, which require an intense draft for efficient combustion. Forced draft is also well adapted for underfeed stokers of the retort type, hollow blast grates, and the closed fire hole system. The air supply may be taken from an air chamber built around the breeching, thereby supplying the heated air to the fan and effecting a lower temperature in the breeching and a higher temperature in the furnace. The objection is sometimes raised against forced draft that the gases tend to pass outward through the fire door when the fire is cleaned or replenished, since the pressure in the furnace is greater than atmospheric. This objection may usually be overcome by suitable dampers in the blast pipe which are closed on opening the fire doors. With a boiler plant of 1000 horse power or more the cost of a forced-draft fan, engine, and stack will approximate from 20 to 30 per cent of the outlay for an equivalent brick chimney. The power consumption will depend upon the character and efficiency of the motor or engine and will range from 1 to 5 per cent of the total capacity.

Induced draft as illustrated in Fig. 172 is perhaps the most common substitute for natural draft and is extensively used in street railway and lighting plants which have high peak loads, being ordinarily installed in connection with fuel economizers. The suction side of the fan is connected with the uptake or breeching of the boiler or batteries of boilers and the products of combustion are usually exhausted through a stub stack. The illustration shows a typical installation in which two fans of the duplex type are placed above the boiler setting. The fan ducts are generally designed with a by-pass direct to the stack to be used in case of accident or when mechanical draft is not required.

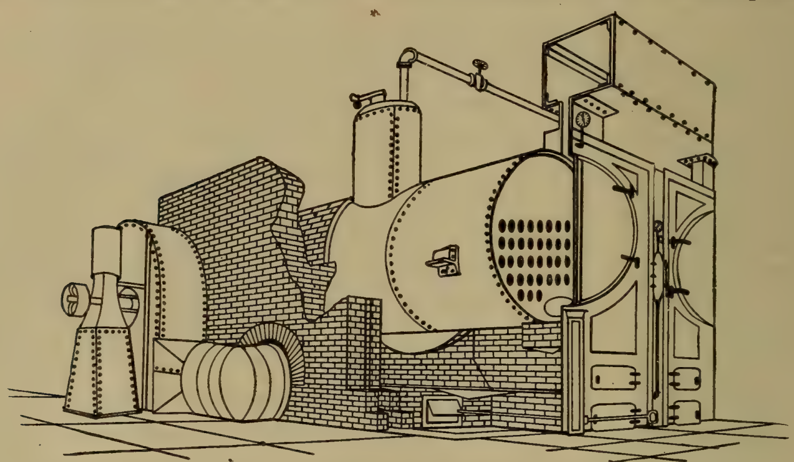


FIG. 171. Typical Forced-draft System.

Since the fan handles hot gases it must, under the ordinary conditions of practice, have a capacity approximately double that of a forced-draft fan delivering cold air, but the gases being of lower density the power required per cubic foot moved is less.

With forced draft about 300 cubic feet of air are required per pound of coal; with induced draft the fan must handle twice this volume if the gases are exhausted at 500 degrees F. or 450 cubic feet if exhausted at 300 degrees F., a temperature to be expected in connection with economizers.

The advantages of induced draft over forced draft are very pronounced. The pressure in the furnace is less than atmospheric, therefore it is not necessary to shut off the draft in cleaning fires or ash pit, and the fire burns more evenly over the entire grate area, since the draft pressures are ordinarily less than with forced draft. An induced-draft plant costs considerably more than forced draft on account of the larger fan required, but the operating expenses are but little greater.

With a boiler plant of 1000 horse power or more the cost of a single induced-draft fan, engine, stack, etc., will approximate from 40 to 50 per cent of the outlay required for a brick chimney of equivalent capacity, and the double-fan outfit will approximate from 50 to 60 per cent. The double-fan system is particularly adapted to plants which operate continuously and where even a temporary break-down is a serious inconvenience.

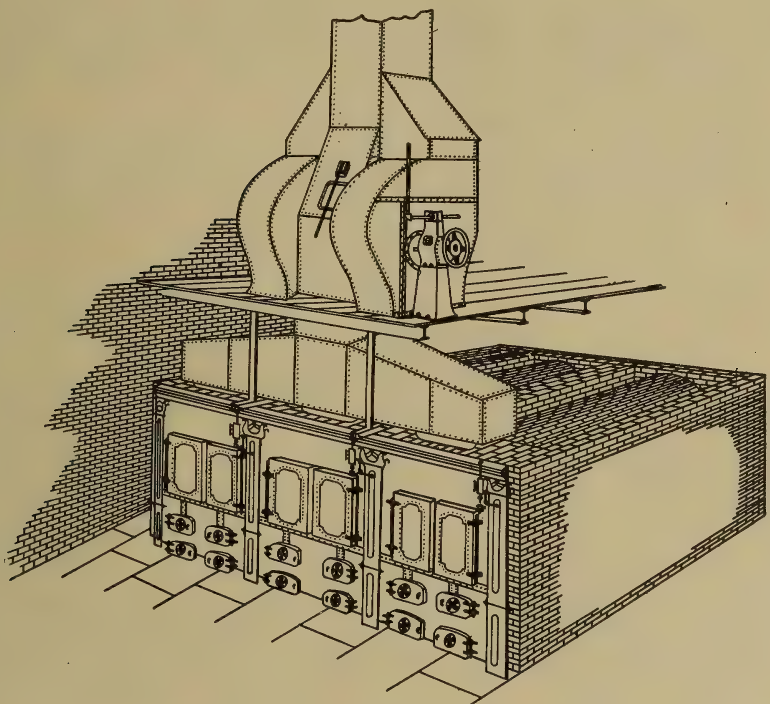


FIG. 172. Typical Induced-draft System.

Turbo-undergrate draft blowers, installed in each setting, are finding favor with many engineers because of the low cost of installation.

They consist essentially of small impulse steam turbines direct connected to specially designed propeller fans set in the side walls of the setting by means of wall thimbles. The fan discharges below the grate, and may be automatically controlled by damper regulation. The turbine exhaust may be discharged into the ash pit to prevent clinkers, or it may be used in the feed-water or other heating devices. They are more economical in heat consumption than the ordinary jet device.

167. Performance of Fans. — The first satisfactory theory of centrifugal fans was promulgated by Daniel Murgue in 1872. He proved that theoretically the maximum pressure created by a perfect fan is equivalent to twice the head which would produce a velocity equal to that of the periphery. Thus

$$H = \frac{u^2}{g}, \tag{82}$$

in which

H = maximum difference in pressure in feet of air,
u = peripheral velocity in feet per second, and
g = acceleration of gravity 32.2.

A and *B*, Fig. 173, represent Pitot tubes inserted in the discharge pipe of a centrifugal blower, *A* being bent to face the current, while *B* is at right angles to it. *A* receives the full impact of the stream, and

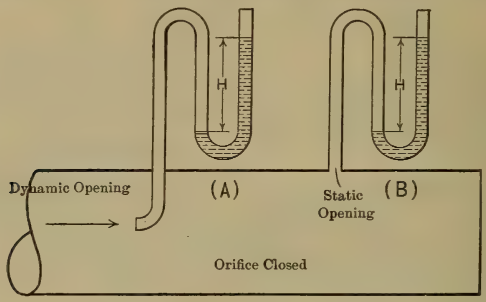


FIG. 173.

the manometer indicates the total pressure, static and velocity, while *B* registers the static pressure only. With the discharge orifice closed, as in Fig. 173, the velocity becomes zero, and the water depression in both manometers will be the same, due to the static pressure, which according to Murgue's theory, will be a maximum and, ignoring friction or eddy currents, $= \frac{u^2}{g}$.

Example: Determine the maximum pressure, in inches of water, which a perfect fan would exert with discharge orifice closed; diameter of fan 6 feet; r.p.m. 318.

The peripheral velocity is

$$u = 2\pi rn = 6.28 \times 3 \times 318 = 6000 \text{ feet per minute}$$
$$= 100 \text{ feet per second.}$$

Substituting in Murgue's formula,

$$u = 100 \text{ and } g = 32.2,$$

$$H = \frac{100^2}{32.2} = 310 \text{ feet,}$$

i.e., the pressure created by the fan would be equivalent to the weight of a column of air 310 feet high, or, assuming an air temperature of 75 degrees F., an equivalent head in inches of water of

$$\frac{310 \times 0.074495}{144 \times 0.0361} = 4.45 \text{ inches.}$$

(0.074495 = density of air at 75 degrees F. and 0.0361 = pressure produced by one inch of water in pounds per square inch.)

If the discharge orifice be opened to its maximum (Fig. 174) the static pressure indicated by manometer B becomes zero, since there is

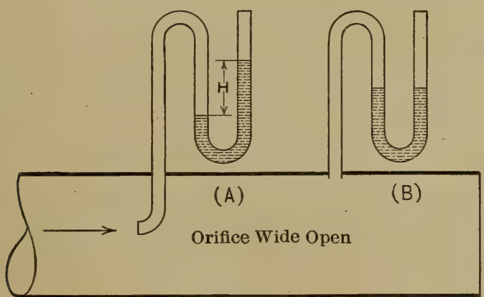


FIG. 174.

no resistance due to the air flow, while the water in A stands at a height H the exact equivalent of the velocity head in accordance with the hydraulic formula,

$$v = \sqrt{2\,gH}, \tag{83}$$

in which v is the velocity of the air in feet per second.

If the orifice be partially closed, say 50 per cent, as in Fig. 175, B indicates the static pressure, while A gives the dynamic or total pressure

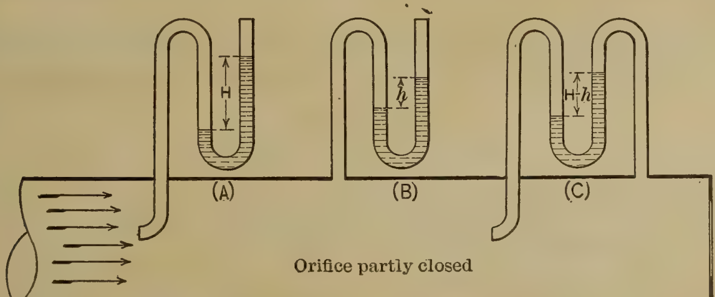


FIG. 175.

due to both velocity and resistance. The difference between A and B is therefore the pressure due to velocity alone. By connecting the

two manometers as indicated in Fig. 175C the velocity pressure is given directly.*

PRESSURE. — According to Murgue's theory the maximum pressure which may be developed by a blower or exhauster varies with the square of the speed and may be expressed

$$p = \frac{C\delta u^2}{g}, \tag{84}$$

in which

p = pressure, pounds per square foot,
δ = density of the air, pounds per cubic foot,
u = peripheral velocity, feet per second,
C = a coefficient obtained by experiment.

Tables 53 and 54 give the relationship between pressure and speed for various sizes of forced and induced-draft fans.

Fig. 178 shows the relationship between pressure and speed in a 45-inch Buffalo blower as tested at the Armour Institute of Technology.

VELOCITY OF DISCHARGE. — The maximum velocity of the air leaving the tips of the blades varies directly as the peripheral speed,

$$V = Ku, \tag{85}$$

in which

V = velocity of the air discharged, feet per second,
K = a coefficient obtained by experiment,
u = peripheral velocity, feet per second.

For practical purposes the velocity of discharge with outlet wide open may be assumed to be that of the periphery.

CAPACITY. — The relationship between capacity and speed, capacity and discharge opening for a 45-inch pressure blower is given in Figs. 177 and 178.

As will be noted, the capacity varies almost directly with the speed of the wheel and the area of discharge as expressed by the equation

$$Q = B\pi ADN, \tag{86}$$

in which

Q = cubic feet discharge per minute,
B = coefficient determined from experiment,
A = area discharge opening, square feet,
D = diameter of the wheel,
N = r.p.m. of the wheel.

* The manometer readings in C, Fig. 175, indicate the velocity pressure for the point D. For a method of determining the *average* velocity of the conduit at a section through D, see Eng. News, Dec. 21, 1905, p. 660.

POWER. — The power required to drive a fan is proportional to the cube of the speed,

$$\text{Horse power} = XAN^3, \tag{87}$$

in which

X = a coefficient determined by experiment,
A = area discharge outlet, square feet,
N = r.p.m.

The marked increase in power required for even a moderate increase in speed should be borne in mind in selecting a fan. (See power curves, Fig. 178.) It is, as a rule, more economical to err in selecting too large a fan than one which must be forced above its rated capacity.

MANOMETRIC EFFICIENCY. — This efficiency is the ratio of the dynamic head as actually observed to the maximum theoretical dynamic head, or

$$E_{\text{man}} = \frac{h}{H}, \tag{88}$$

in which h is determined from the actual manometer reading and H is calculated from equation (82).

VOLUMETRIC EFFICIENCY. — This is the ratio between the actual volume of air passing in a given time divided by the impeller displacement for the same period, or

$$E_{\text{vol}} = \frac{4\,Q}{\pi D^2 NB}, \tag{89}$$

in which

Q = volume discharged, cubic feet per minute,
D = diameter of the impeller, feet,
B = width of the impeller, feet,
N = r.p.m.

MECHANICAL EFFICIENCY, or simply fan efficiency is the ratio of the total work done by the fan in moving the air to the horse-power input to the fan, or

$$E_{\text{mec}} = \frac{Q'h}{Hi \times 33{,}000}, \tag{90}$$

in which

Q' = weight discharged, pounds per minute,
h = dynamic head, feet of air,
Hi = horse-power input.

In practice the size of fan is proportioned upon experience rather than theory, the usual procedure necessitating the use of curves based upon the performance of fans of the type under consideration.

The curves in Fig. 176 were computed by Mr. F. R. Still of the American Blower Company, and give the performance of steel-plate fans as manufactured by this company. These curves apply to this type and make of fan only, though the difference is not very great for any type of centrifugal fan. The "ratio of opening" refers to the actual percentage of opening compared with the total discharge. The

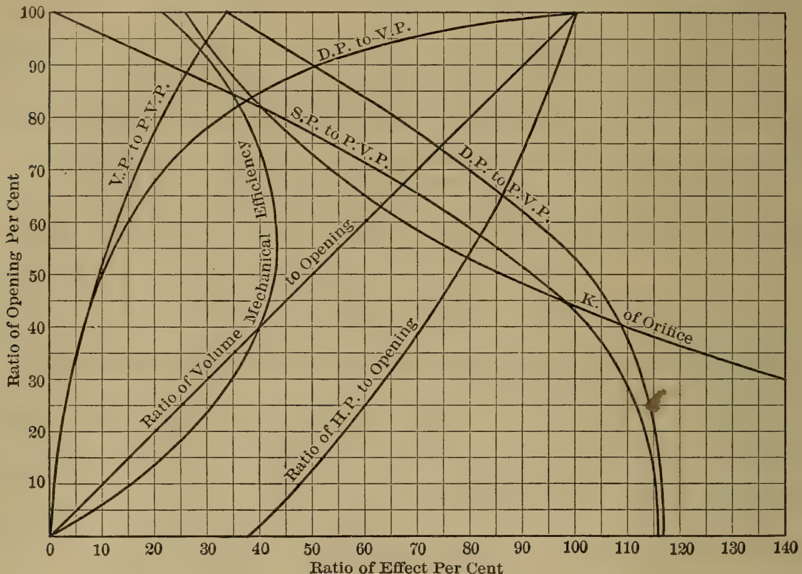


FIG. 176. Performance of Steel Plate Fans.

"ratio of effect" is the relative effect produced by restricting the discharge. The abbreviations are as follows:

D.P. = dynamic or total pressure.

P.V.P. = pressure created by a column of air moving at the same velocity as the periphery.

S.P. = static pressure.

V.P. = velocity pressure = D.P. − S.P.

Suppose a fan with an unrestricted inlet and outlet delivers 25,000 cubic feet of air per minute against a head (D.P.) of 0.33 inch with a peripheral velocity requiring 6.16 horse power. It appears from the curves that if the discharge outlet is restricted to 50 per cent of the full area, only 12,500 cubic feet will be delivered; the pressure will be increased to 1.03 inches, and the power required drops to 4.84 horse power. If the outlet be still further reduced to 20 per cent of the full opening the capacity will drop to 5000 cubic feet, the pressure will increase to 1.15 inches, and the power will be decreased to 3.45 horse

power. With a discharge area of 60 per cent, the mechanical efficiency is a maximum, and equal to about 43 per cent. With orifice closed the horse power required to drive the fan is about 37 per cent of that required when discharging the maximum volume of air.

Curve "K" in Fig. 176 was determined from the empirical formula (based upon Murgue's theorem)

$$A = \frac{KQ}{\sqrt{P}}, \tag{90a}$$

in which

A = area of the inlet orifice, square feet,
Q = volume of gas, thousands of cubic feet per minute,
P = draft at the inlet in inches of water,
K = constant determined by experiment.

The curves in Figs. 177 and 178 are plotted from tests made at the Armour Institute of Technology on a 45-inch Buffalo pressure blower, and are characteristic of this type of fan.

Measurement of Air in Fan Work: C. H. Treat, Jour. A.S.M.E., Sept., 1912, p. 1341. *Some Experiences with the Pitot Tube on High and Low Air Velocities:* F. H. Kneeland, Jour. A.S.M.E., Nov., 1911, p. 1407. *Experiments with Ventilating Fans and Pipes:* Capt. D. W. Taylor, Soc. Naval Arch. and Marine Engrs., 1905, p. 35. *The Measurements of Gases:* Carl C. Thomas, Jour. Frank. Inst., Nov., 1911, p. 411. *Experiments with the Pitot Tube in Measuring the Velocities of Gases:* R. Burnham, Eng. News, Dec. 21, 1905, p. 660. *Pressure Fans vs. Exhaust Fan:* Bulletin Am. Inst. Min. Engrs., Feb., 1909.

168. Determination of Size of Fan. — The following analysis, based upon a paper on Mechanical Draft by F. R. Still of the American Blower Company, gives a good idea of the usual procedure in determining the size of fan for an induced draft installation. (Jour. West. Soc. Engr., May, 1902.)

Example: Determine the size of induced fan and the approximate power required to drive it, for a boiler plant rated at 1000 horse power; temperature of flue gases 500 degrees F.; heat value of coal 14,000 B.t.u. per pound; ash 5 per cent; draft required, 1 inch of water pressure.

Assuming a boiler efficiency of 70 per cent, the evaporation will be $\frac{14,000}{966} \times 0.70 = 10.15$ pounds of water from and at 212 degrees F. per pound of coal.

Since one boiler horse power is equivalent to the evaporation of 34.5 pounds of water per hour from and at 212 degrees F., the evaporation per hour will be 34.5 × 1000 = 34,500 pounds, and the coal burned per hour,

$$\frac{34,500}{10.15} = 3400 \text{ pounds.}$$

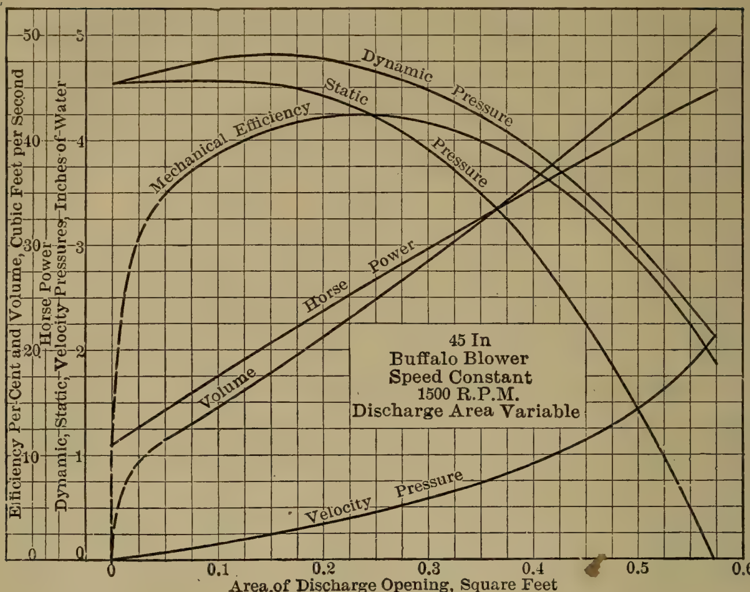


FIG. 177.

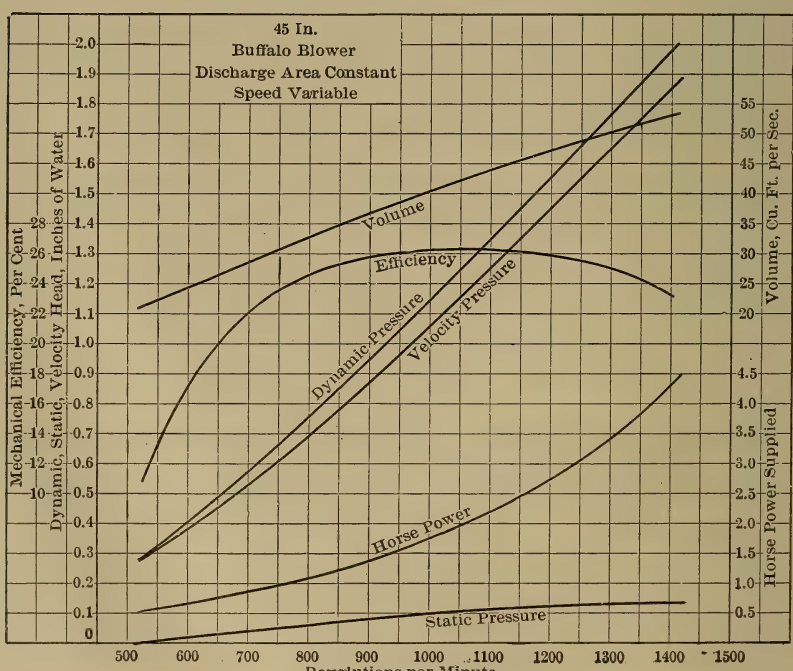


FIG. 178.

Allowing 18 pounds of flue gas per pound of combustible, 5 per cent for ash and 5 per cent for leaks, the fan will have to handle, at 500 degrees F., approximately 20 × 3400 = 68,000 pounds of gas per hour, or 26,000 cubic feet per minute. It is customary, when little is known about a plant in which a fan is to be installed, to assume that the resistance is equivalent to restricting the discharge outlet 25 per cent. Hence, in this problem the various factors are referred to a "ratio of opening" of 75 per cent (see Fig. 176).

From formula 90a, the area of the inlet should be

$$A = \frac{KQ}{\sqrt{P}} = \frac{0.485 \times 26}{1} = 12.6 \text{ square feet,}$$

which corresponds to a diameter of 48 inches. ($K = 0.485$ is taken from the curves in Fig. 176.)

The area of the inlet should not exceed 40 per cent of the area of the side of the wheel; the latter, then, will be

$$\frac{12.6}{0.4} = 31.5 \text{ square feet,}$$

which corresponds to a diameter of 76 inches (6.3 feet).

Referring to Fig. 176, the ratio of dynamic pressure to peripheral velocity pressure (D.P. to P.V.P. at 75 per cent opening) is 0.73. The peripheral velocity pressure will be $\frac{1}{0.73} = 1.37$ inches of water.

The peripheral velocity is

$U = \sqrt{2\,gH'} = 8.03\,\sqrt{H'}$, where H' is the peripheral velocity pressure expressed in feet of gas, or,

since $$H' = \frac{p \times 62.4}{0.0435 \times 12}, \text{ where } p = \text{inches water,}$$

$$U = 87.5\,\sqrt{1.37}$$
$$= 102.5 \text{ feet per second}$$
$$= 6150 \text{ feet per minute.}$$

The maximum effective discharge area, which an inclosed fan of this type may have, and still maintain the pressure equivalent of the peripheral velocity, is usually called the "blast area." With a larger area the pressure will be reduced, but with a smaller area will remain substantially constant. The velocity of the discharge is practically that of the tips of the blades, whence the blast area is equal to $\frac{26,000}{6150} = 4.23$ square feet, which, with this type of fan, is found to be about $\frac{1}{3}$ the projected rectangle of the wheel, therefore,

The projected rectangle = 4.23 × 3 = 12.7 square feet.

The proper width of periphery is found by dividing this area by the wheel diameter, thus,

$$\text{width of blades} = \frac{12.7}{6.3} = 2.02 \text{ feet} = 24.2 \text{ inches},$$

and

$$\text{speed of fan} = \frac{6150}{3.14 \times 6.3} = 311 \text{ r.p.m.}$$

$$W = \frac{\text{Volume of gas (cu. ft. per min.)} \times \text{Pressure (lb. per sq. ft.)}}{33{,}000 \times \text{efficiency of fan}}.$$

$$W = \frac{26{,}000 \times 5.2}{33{,}000 \times 0.4} = 10.2 \text{ brake horse power.}$$

(5.2 = pressure in pounds per square foot equivalent to one inch of water, and 0.4 is the mechanical efficiency for 75 per cent opening as taken from curve in Fig. 176.)

Assuming a steam consumption of 70 pounds per brake horse power for a small, simple non-condensing high-speed engine, the steam consumed per hour will be

$10.2 \times 70 = 714$ pounds per hour, or 2.3 per cent of the total steam capacity of the boilers.

Table 53 gives the capacity and horse power required for various sizes of forced-draft fans, and Table 54 gives similar data for induced-draft fans.

169. Chimney vs. Mechanical Draft. — The choice of chimney or mechanical draft depends largely upon local conditions. Many power plants with tall stacks are provided with forced-draft apparatus to be used in emergencies, but, as a general rule, where ordinances require high chimneys mechanical draft is not considered. In a few isolated cases stokers of the forced-draft type are used in connection with chimneys as high as 250 feet, but such installations are limited to large central stations with heavy peak loads.

Where there are no limitations to the height of stack, mechanical draft offers many advantages over chimney draft, especially for railroad work and large lighting plants. With certain types of grates and for low-grade fuels and anthracite culm or dust, it is indispensable. Again, where a fair quality of fuel is obtainable the size of plant may determine the choice.

First Cost: In small plants of, say, 100 to 150 horse power the cost of a guyed steel chimney, 75 feet in height or less, would be considerably less than that of a mechanical-draft system, and once erected would cost practically nothing for operation, while the power required to operate a fan in so small a plant would amount to 5 per cent or more of the total steaming capacity.

TABLE 53.

CAPACITIES OF FORCED-DRAFT FANS.

(Steel Plate Fans.)

For Forced Draft, Temperature of Air 60°.

Diameter of Fan.	Cubic Feet of Air Delivered to Furnace per Minute.	Pressure in Inches of Water.													
		0.5		0.75		1.00		1.25		1.50		2.00		2.50	
		R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.
2′ 6″	4,200	510	1.6	560	1.8	600	1.9	640	2.1	710	2.3	780	2.5	850	2.7
3′	5,800	430	2.2	460	2.4	490	2.6	530	2.8	590	3.1	640	3.4	710	3.8
3′ 6″	7,800	360	3.0	400	3.3	420	3.5	450	3.8	500	4.2	550	4.6	610	5.1
4′	10,000	320	3.9	350	4.2	370	4.4	400	4.9	440	5.4	480	5.9	530	6.5
4′ 6″	12,400	290	4.8	310	5.2	330	5.6	360	6.0	400	6.7	430	7.3	470	8.0
5′	15,200	250	5.9	270	6.4	290	6.8	310	7.4	350	8.2	380	8.9	420	9.8
5′ 6″	18,200	230	7.0	250	7.7	270	8.2	300	8.8	330	9.8	360	10.6	390	11.8
6′	21,400	210	8.3	230	9.1	250	9.6	260	10.4	290	11.5	320	12.5	350	13.9
7′	28,800	180	11.2	200	12.2	210	13.0	230	14.0	250	15.5	280	16.8	300	18.7
8′	37,200	160	14.4	170	15.7	190	16.7	200	18.1	220	20.1	240	21.8	270	22.5
9′	46,800	140	18.1	160	19.8	170	21.1	180	22.7	200	25.3	220	27.4	240	30.3
10′	57,400	130	22.2	140	24.3	150	25.8	160	27.9	180	3.1	200	33.6	210	37.2

Discharge velocity 2000 feet per minute.

TABLE 54.

CAPACITIES OF INDUCED-DRAFT FANS.

(Steel Plate Fans.)

For Induced Draft, Temp. of Flue Gases 500°.

Diameter of Fan.	Cubic Feet of Air at 60° Temp. Drawn into Furnace per Minute.	Pressure in Inches of Water.													
		0.5		0.75		1.00		1.25		1.50		2.00		2.50	
		R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.
2′ 6″	3,000	688	2.2	756	2.4	810	2.6	864	2.8	958	3.1	1053	3.4	1147	3.6
3′	4,200	580	3.0	621	3.2	661	3.5	715	3.8	796	4.2	864	4.6	958	5.1
3′ 6″	5,700	486	4.0	540	4.5	567	4.7	607	5.1	675	5.7	742	6.2	823	6.9
4′	7,300	432	5.3	472	5.7	500	6.1	540	6.6	594	7.3	648	8.0	715	8.8
4′ 6″	9,300	390	6.5	418	7.0	445	7.5	486	8.1	540	9.0	580	9.8	634	10.8
5′	11,100	337	8.0	364	8.6	391	9.2	418	10.0	472	11.1	513	12.0	567	13.2
5′ 6″	13,300	310	9.5	337	10.4	364	11.1	405	11.9	445	13.2	486	14.3	526	15.9
6′	15,600	283	11.2	310	12.3	337	13.0	351	14.0	391	15.5	432	16.9	472	18.7
7′	21,000	243	15.1	270	16.5	283	17.5	310	18.9	337	20.9	378	22.6	405	25.2
8′	27,100	216	19.4	230	21.2	256	22.5	270	24.4	297	27.1	324	29.4	364	30.4
9′	34,200	189	24.4	216	26.7	230	28.5	243	30.6	270	34.1	297	37.0	324	40.9
10′	41,900	175	30.0	190	32.8	202	34.8	216	37.6	243	41.8	270	45.3	283	50.2

TABLE 55.

CAPACITIES OF FORCED-DRAFT FANS.*

(Sirocco Type.)

(Figures given Represent Dynamic Pressures in Ozs. per Sq. In. For Static Pressure Deduct 28.8 Per Cent. For Velocity Pressure Deduct 71.2 Per Cent.)

Diam. Wheel.		¼ Oz.	½ Oz.	¾ Oz.	1 Oz.	1¼ Oz.	1½ Oz.	1¾ Oz.	2 Oz.	2½ Oz.	3 Oz.
	Cu. Ft.	155	220	270	310	350	380	410	440	490	540
6	R.P.M.	1,145	1,615	1,980	2,290	2,560	2.800	3,025	3,230	3,616	3,960
	B.H.P.	0.0185	0.052	0.095	0.147	0.205	0.270	0.34	0.42	0.58	0.76
	Cu. Ft.	625	880	1,080	1,250	1,400	1,530	1,650	1,770	1,970	2,170
12	R.P.M.	572	808	990	1,145	1,280	1,400	1,512	1,615	1,808	1,980
	B.H.P.	0.074	0.208	0.381	0.588	0.82	1.08	1.36	1.66	2.32	3.05
	Cu. Ft.	1,410	1,990	2,440	2,820	3,160	3,450	3,720	3,980	4,450	4,880
18	R.P.M.	381	538	660	762	850	933	1,010	1,076	1,204	1,320
	B.H.P.	0.167	0.470	0.862	1.33	1.85	2.43	3.07	3.75	5.25	6.9
	Cu. Ft.	2,500	3,540	4,340	5,000	5,600	6,120	6,620	7,080	7,900	8,680
24	R.P.M.	286	404	495	572	640	700	756	807	904	990
	B.H.P.	0.296	0.832	1.53	2.35	3.28	4.32	5.44	6.64	9.3	12.2
	Cu. Ft.	3,910	5,520	6,770	7,820	8,750	9,600	10,350	11,050	12,350	13,550
30	R.P.M.	228	322	395	456	510	560	604	645	722	790
	B.H.P.	0.460	1.30	2.40	3.68	5.15	6.75	8.53	10.4	14.5	19.1
	Cu. Ft.	5,650	7,950	9,750	11,300	12,640	13,800	14,900	15,900	17,800	19,500
36	R.P.M.	190	269	330	381	425	466	504	538	602	660
	B.H.P.	0.665	1.87	3.44	5.30	7.40	9.72	12.25	15.0	20.9	27.5
	Cu. Ft.	10,000	14,150	17,350	20,000	22,400	24,500	26,500	28,300	31,600	34,700
48	R.P.M.	143	202	248	286	320	350	378	403	452	495
	B.H.P.	1.18	3.32	6.10	9.40	13.1	17.2	21.75	26.6	37.1	48.8
	Cu. Ft.	15,650	22,100	27,100	31,300	35,000	38,400	41,400	44,200	49,400	54,200
60	R.P.M.	114	161	198	228	255	280	302	322	361	396
	B.H.P.	1.84	5.20	9.58	14.7	20.6	27.0	34.1	41.6	58.2	76.5
	Cu. Ft.	22,600	31,800	39,000	45,200	50,600	55,200	59,600	63,600	71,200	78,000
72	R.P.M.	95	134	165	190	212	233	252	269	301	330
	B.H.P.	2.66	7.48	13.7	21.2	29.6	33.9	49.0	59.8	83.6	110
	Cu. Ft.	30,800	43,400	53,200	61,600	68,700	75,200	81,200	86,800	97,100	106,400
84	R.P.M.	81	115	142	163	182	200	216	231	258	283
	B.H.P.	3.61	10.2	18.7	28.9	40.4	53.0	66.8	81.7	114	150
	Cu. Ft.	35,250	49,800	61,000	70,500	78,800	86,400	93,300	99,600	111,200	122,000
90	R.P.M.	76	107	132	152	170	186	201	214	241	264
	B.H.P.	4.14	11.7	21.5	33.1	46.2	60.7	76.7	93.6	131	172

* A number of sizes have been omitted.

A tall, self-supporting chimney for larger plants, however, is very costly as compared with a fan system of equal capacity. For example, a brick chimney 175 feet high and 10 feet in diameter, foundation and all, capable of furnishing the necessary draft for a 3000-horse-power plant, will cost about $10,000. A two-fan induced system of equivalent capacity will cost in the neighborhood of $5000, a one-fan system $3500, and a forced-draft system $2500. See Fig. 179. With interest at 5 per cent, depreciation 5 per cent, taxes 1 per cent, and insurance one-half per cent, the annual fixed charges will be $575, $402.50, $287.50 respectively, for the fan equipment.

Depreciation and Maintenance: The depreciation of a well-designed masonry or concrete stack is very low, and 2 per cent is a liberal factor. Maintenance is practically negligible, as it requires no attention whatever for years. A steel stack, however, must be kept well painted or corrosion will take place rapidly. The depreciation and maintenance charges on a mechanical-draft system will range from 4 per cent to 10 per cent of the original outlay.

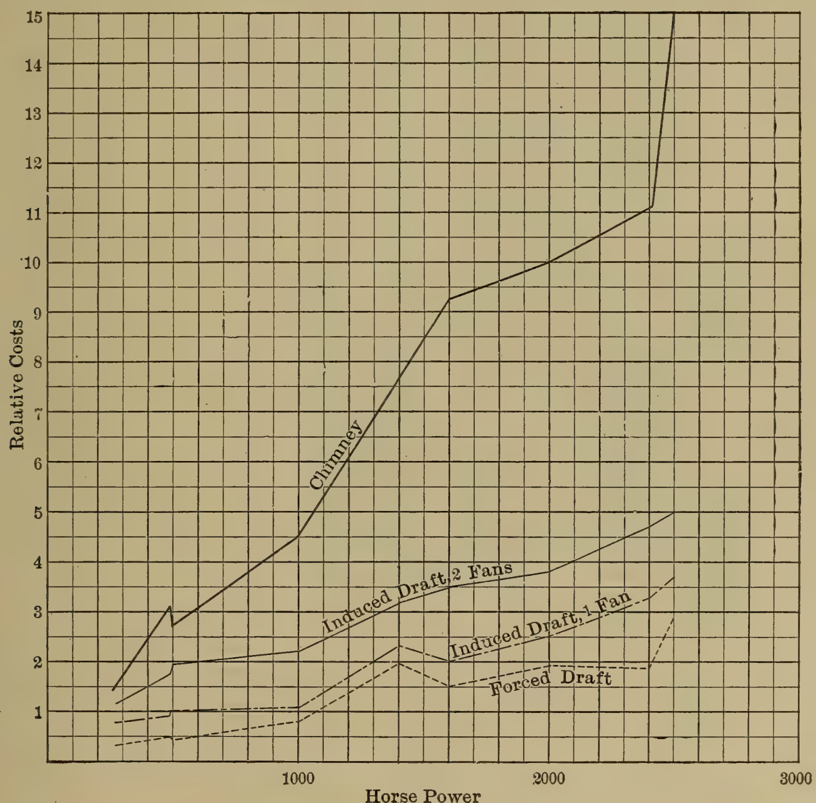


FIG. 179. Comparative Costs of Chimneys and Mechanical Draft. (W. B. Snow.)

Cost of Operation: Once erected, the comparative cost of operating a chimney is practically nothing; that is, of course, on the assumption that the chimney and fan exhaust equal volumes of gas per pound of fuel and at the same temperature. A fan system requires for its operation from one and one-half per cent to five per cent of the total steaming capacity of the plant, depending upon the type and character of the fan engine or motor, and the conditions of operation.

Efficiency: With fan draft a very thick fire can be maintained on the grate, thus permitting a high rate of combustion, and minimum air per pound of fuel, both of which result in increased boiler efficiency. The influence of the rate of combustion on air supply in a specific case is illustrated in Fig. 180. For the same temperature of discharge each pound of air in excess of theoretical requirements results in a loss of about one per cent of the total heat in the fuel. With fan draft an average figure is 18 pounds of air per pound of bituminous coal against 24 pounds for the chimney, a saving of 5 per cent in favor of the fan. Again, a fan permits of a low temperature of the flue gases without affecting the draft, while lowering the temperature in the chimney

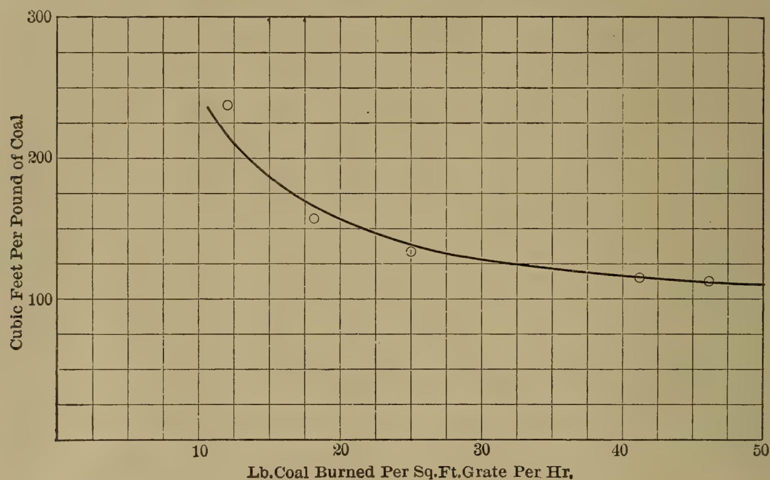


FIG. 180. Influence of Rate of Combustion on Air Supply. — Forced Draft.

reduces the draft as shown in Table 42. From Table 11 we see that a reduction in flue gas temperature of 25 degrees F. will increase the boiler efficiency about one per cent. With an economizer the flue gases may be reduced to 350 degrees F., with a net saving of about 500 − 350 = 150, or 6 per cent of the total fuel. It is in this connection that the fan draft is peculiarly suitable. Of course, the chimney may be provided with an economizer, effecting the same reduction in temperature, but its height must be made sufficiently great to overcome the additional resistance of the economizer and the reduction in temperature of the chimney gases.

Flexibility: With a fan the draft may be readily regulated for sudden increased or decreased requirements, independent of the boiler performance. Damp and muggy days appreciably affect the draft of a chimney, as do adverse air currents and high winds.

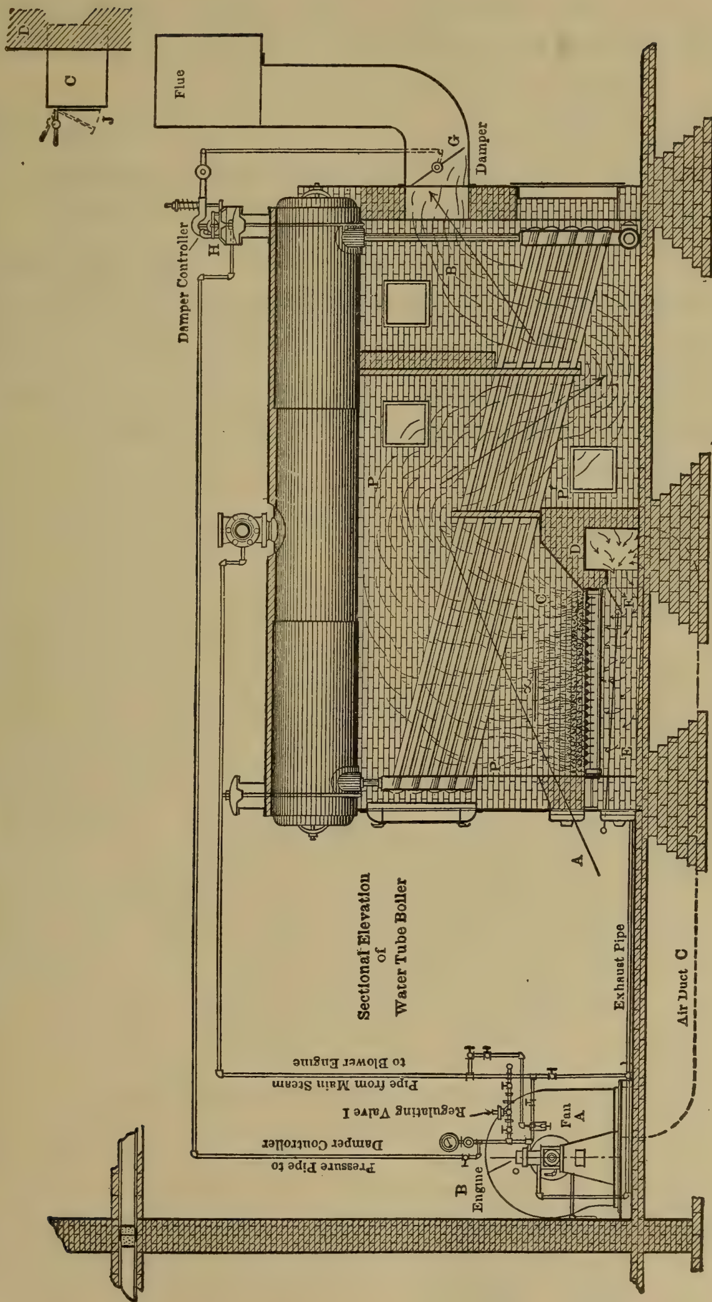


Fig. 181. "Balanced Draft" System.

Smoke: Smokeless combustion is more readily effected with artificial draft than with natural draft, as a thicker fire can be carried, and the correct proportion of air can be more readily adjusted.

170. Balanced Draft. — Fig. 181 illustrates an application of the McLean "Balanced Draft" system to a water-tube boiler. The equipment consists of a blower, the speed of which is regulated by the steam pressure, so that the draft in the fire box is maintained at approximately atmospheric pressure. The chief claims for this system are: (1) the velocity of the gases over the tubes is reduced, and short circuiting is prevented; (2) the correct proportion of air to fuel is readily maintained; (3) infiltration of air through the setting is impossible, as the pressures are "balanced"; (4) sudden changes in load are correctly taken care of. Tests of the apparatus at the Fuller Building, New York, gave excellent results (Trans. A.S.M.E., 26–641).

CHAPTER IX.

RECIPROCATING STEAM ENGINES.

171. Introductory. — The type of prime mover best suited [for a given installation is the one which delivers the required power at the lowest cost, taking into consideration all charges, fixed and operating. These include not only the cost of fuel, labor, supplies and repairs, but all overhead charges such as interest on the investment, depreciation, maintenance and taxes. Space requirements and continuity of operation are often of vital importance, and may greatly influence the selection of type of prime mover and auxiliary apparatus. In many situations the gas engine and producer are productive of the highest commercial economy; in others the choice lies between the reciprocating steam engine or turbine, occasionally the hydroelectric plant offers the best returns, but in general each proposed installation is a problem in itself, and general rules are without purpose.

The reciprocating steam engine is the most widely distributed prime mover in the power world, and although its field of usefulness has been greatly encroached upon in recent years by the steam turbine and gas engine it is still an important heat engine and will probably continue to be a factor for years to come. In a general sense the piston engine is superior to the turbine for variable speed, slow rotative speeds and heavy starting torque, while the turbine has practically superseded the engine for large central station units and for auxiliaries requiring high rotative speed. From a purely thermal standpoint the internal combustion engine is vastly superior to the steam engine and the turbine is more economical in space requirements, but taking into consideration all of the items affecting the production of power, the reciprocating engine may still prove to be the better investment in many situations.

172. The Ideal Engine. — In every heat engine the working fluid goes through a circuit or *cycle* of operation. Beginning at a particular condition it passes through a series of successive states of pressure, volume and temperature and returns to the initial condition. An ideally perfect engine which effects the highest possible conversion of heat into mechanical work for a given cycle is taken as a standard of comparison for the performance of the actual engine. Two such standards are adopted in connection with the steam engine, (1) the

ideal engine operating in the Carnot cycle, and (2) the ideal engine operating in the Rankine cycle.

173. The Carnot Cycle. — The diagrams in Fig. 182 represent the action of the working fluid in an ideal steam engine cylinder, operating in the Carnot cycle. (*A*) represents the familiar *indicator card* or pressure-volume diagram, and (*B*) the *temperature-entropy* diagram. The former illustrates the kinetic action of the steam in the cylinder and the latter the thermal action. At the beginning of the stroke, *a*, the non-conducting cylinder contains a mixture of steam and water at temperature T_1 and pressure p_1. Heat is applied at temperature T_1 and pressure p_1 until a part or all of the liquid is vaporized and the isothermal expansion forces the frictionless piston to position *b*. From

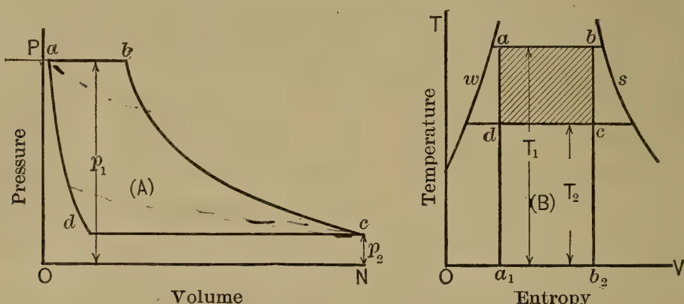


Fig. 182. The Carnot Cycle (Saturated Steam).

b to *c* the vapor expands adiabatically and forces the piston to the end of its stroke. On the return stroke from *c* to *d* the working fluid is compressed isothermally with condensation and rapid rejection of heat. The cycle is completed by an adiabatic compression from *d* to *a*.*

If x = the quality of the steam at the point indicated by the subscript,

r = latent heat of vaporization at the pressure indicated by the subscript, B.t.u. per pound of steam,

T = absolute temperature of the steam, degrees F.

The heat H_1 absorbed by one pound of the mixture in passing from *a* to *b* is

$$H_1 = r_1 x_b - r_1 x_a. \tag{91}$$

The heat H_2 rejected in passing from *c* to *d* is

$$H_2 = r_2 x_c - r_2 x_d. \tag{92}$$

* It is commonly assumed that there is only water in the cylinder at *a* and saturated steam at *b*, but there is no necessity for such an assumption, and it in no way affects the efficiency.

The heat H_c converted into work during the cycle is

$$\begin{aligned} H_c &= H_1 - H_2 \\ &= r_1(x_b - x_a) - r_2(x_c - x_d). \end{aligned} \tag{93}$$

From thermodynamics we have as the adiabatic equation for a liquid and its vapor

$$r_2(x_c - x_d) = \frac{T_2}{T_1} r_1(x_b - x_a). \tag{94}$$

Substituting this value of $r_2(x_c - x_d)$ in equation (93) we get

$$H_c = r_1(x_b - x_a)\frac{T_1 - T_2}{T_1}. \tag{95}$$

The efficiency E_c of the perfect engine operating in the Carnot cycle is

$$E_c = \frac{H_1 - H_2}{H_1} = \frac{r_1(x_b - x_a)\dfrac{T_1 - T_2}{T_1}}{r_1x_b - r_1x_a} = \frac{T_1 - T_2}{T_1}, \tag{96}$$

which is independent of the nature of the working substance and dependent only on the range in temperature. The upper limit of temperature is that corresponding to boiler pressure, and the lower limit to that of the exhaust steam. Evidently the greater this temperature range the more nearly does this efficiency approach unity, but with the present limits of temperature used in steam engines it cannot exceed 38 per cent.

In the wholly ideal Carnot cycle the entire cycle — heat reception and expansion, heat rejection and compression — is supposed to be performed within the cylinder itself, using an unchanged body of working substance over and over again. While not absolutely impossible this manner of operation is commercially impracticable.

The nearest approach of any actual engine to the Carnot cycle is accomplished by the Nordberg system of progressive feed-water heating, in which the water is successively heated from the receivers intermediate between each pair of cylinders. (For a description of this engine see Eng. News, May 4, 1899, p. 283.)

Example: Determine the efficiency of the ideal engine working in the Carnot cycle if the boiler pressure is 200 pounds absolute and the back pressure 2 pounds absolute.

$$\begin{aligned} T_1 &= 388 + 459.6 = 847.6. \\ T_2 &= 126.1 + 459.6 = 585.7. \\ E &= \frac{847.6 - 585.7}{847.6} \\ &= 0.309 \text{ or } 30.9 \text{ per cent.} \end{aligned}$$

174. The Rankine Cycle with Complete Expansion.* — The Carnot cycle is practically impossible for an engine using superheated steam at constant pressure, and in general is not closely simulated by an engine using saturated steam. It represents, however, the theoretical limit of perfection of any heat engine.

The diagrams in Fig. 183 represent the action of the working fluid in an ideal engine cylinder, operating in the Rankine cycle, which closely parallels the cycle of the actual engine. *ab* represents the admission of steam from the boiler at pressure p_1; *bc* is an adiabatic expansion to exhaust pressure p_2; *cd* represents the exhaust, and *da* is an adiabatic compression to the initial pressure.

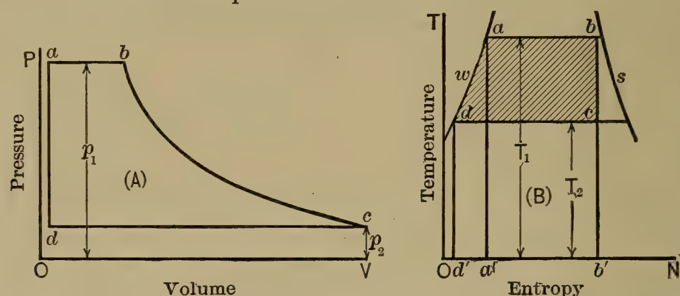


FIG. 183. The Rankine Cycle with Complete Expansion (Saturated Steam).

Let H_1 = the total heat of one pound of steam at pressure p_1.

H_2 = total heat of one pound of steam at pressure p_2 after adiabatic expansion from pressure p_1.

q_2 = heat of the liquid at pressure p_2.

The heat changed into work per pound of steam is

$$H_1 - H_2. \tag{97}$$

The heat necessary to raise the feed water from the temperature of exhaust to the temperature in the boiler and evaporate it is

$$H_1 - q_2. \tag{98}$$

The efficiency, E_r, of the cycle, or the ratio of the heat equivalent of the useful work to the heat supplied, is

$$E_r = \frac{H_1 - H_2}{H_1 - q_2}, \tag{99}$$

* This is often called the Clausius cycle since it was published simultaneously, but independently, by both Clausius and Rankine. This cycle has been adopted by the British Society of Civil Engineers, and is generally accepted in this country as the standard of comparison for steam engines and turbines.

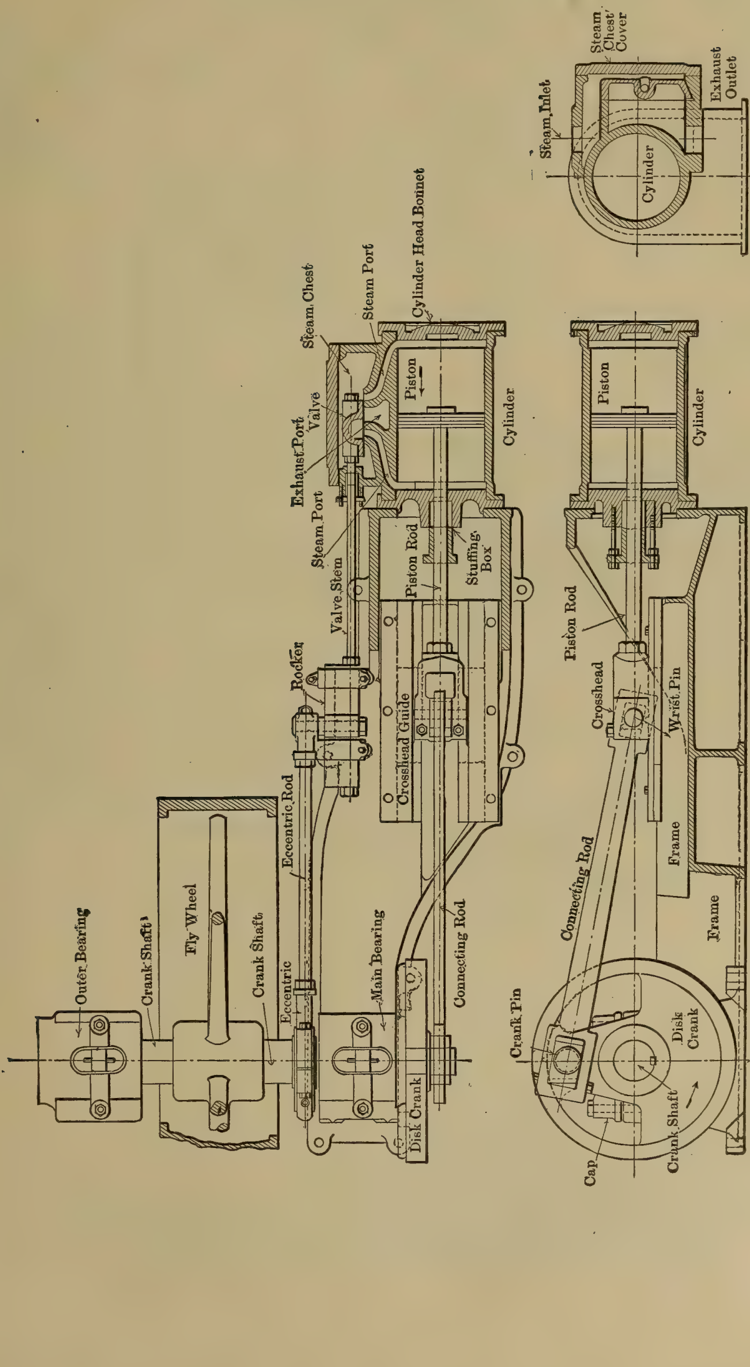


FIG. 184. Typical Piston Engine, Single Cylinder, Throttling Governor.

and the water rate of steam consumption of the perfect engine, W_r, pounds per horse-power hour, may be expressed as

$$W_r = \frac{2546}{H_1 - H_2}. \tag{100}$$

For dry steam $H_1 = r_1 + q_1$, (101)

For wet steam $H_1 = x_1 r_1, + q_1$, (102)

For superheated steam $H_1 = r_1 + q_1 + C_1 t_s$, (103)

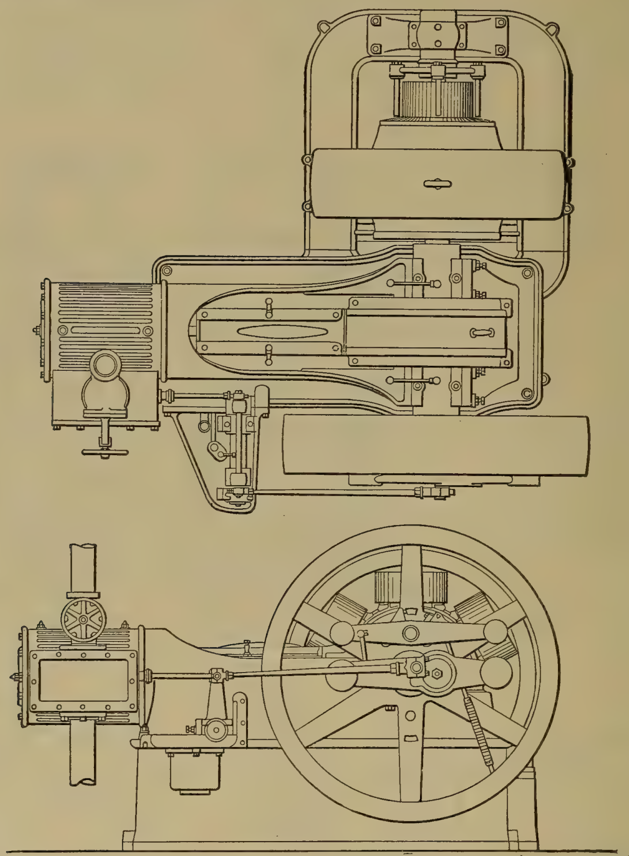


Fig. 185. Typical Piston Engine, Single Cylinder, Automatic Governor.

in which

r_1 = heat of vaporization at pressure p_1,

x_1 = quality of steam at pressure p_1,

q_1 = heat of the liquid at pressure p_1,

C_1 = mean specific heat of the superheated steam at pressure p_1,

t_s = degree of superheat or the difference in temperature between the superheated and saturated steam at pressure p_1.

If the steam after adiabatic expansion is wet as is the usual case

$$H_2 = x_2 r_2 + q_2. \tag{104}$$

If initial superheat is so high that the steam at the end of expansion is still superheated

$$H_2 = r_2 + q_2 + C' t_s', \tag{105}$$

in which

x_2, r_2, q_2 = quality, heat of vaporization and heat of the liquid respectively, at pressure p_2,

C' = mean specific heat of superheated steam at pressure p_2,

t_s' = degree of superheat at pressure p_2.

Before the *total heat-entropy* or *Mollier diagram* came into common use it was necessary to calculate H_2, x_2, and C' from thermodynamic equations, a tedious and laborious procedure and particularly so with highly superheated steam. With the aid of this diagram all problems involving adiabatic expansion are solved with ease and accuracy; in fact, the Mollier diagram has to all intents and purposes supplanted the steam tables in this connection. For this reason, the thermodynamic equations for solving H_2, x_2, and C' will be omitted. See appendix L.

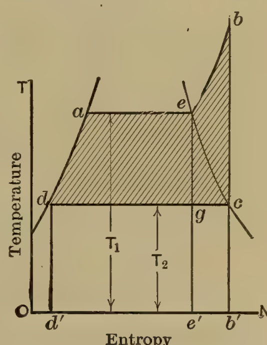


FIG. 186. The Rankine Cycle for Superheated Steam.

Example: Determine the efficiency and water rate of the ideal engine working in the Rankine cycle if the steam pressure is 200 pounds absolute, superheat 250 degrees F., and exhaust pressure 0.5 pounds absolute.

From steam tables or the Mollier diagram we find

$$H_1 = 1332 \text{ B.t.u. and } q_2 = 48 \text{ B.t.u.},$$

From the Mollier diagram we find

$$H_2 = 908 \text{ B.t.u.},$$

$$E_r = \frac{1332 - 908}{1332 - 48} = 0.33 \text{ or } 33 \text{ per cent},$$

$$W_r = \frac{2546}{1332 - 908} = 6.0 \text{ pounds per horse-power hour.}$$

175. The Rankine Cycle with Incomplete Expansion. — If expansion after cut-off is not carried far enough to reduce the pressure to that of the back pressure line, as shown in Fig. 187, the Rankine cycle more nearly simulates the cycle of the actual engine. This cutting off the "toe" of the diagram decreases the ideal efficiency, but permits of the use of a smaller cylinder.

The work during admission is

$$W_1 = p_1 (x_1 u_1 + \sigma) \text{ foot-pounds,} \quad (106)$$

in which

u_1 = increase of volume due to vaporization of a pound of steam,

σ = specific volume of water.

The work during expansion from b to c is

$$W_2 = (x_1 \rho_1 + q_1 - x_c \rho_c - q_c) \times 778 \text{ foot-pounds,} \quad (107)$$

in which

x, ρ, and q are the quality, heat equivalent of the internal work during vaporization and the heat of the liquid respectively at pressures indicated by the subscript.

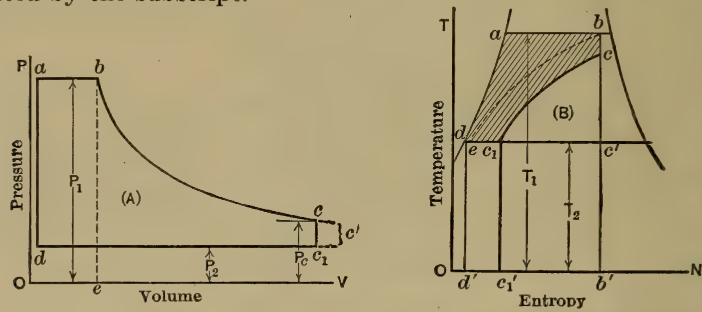


FIG. 187. The Rankine Cycle with Incomplete Expansion (Saturated Steam).

The work done by the piston on the steam during exhaust is

$$W_3 = p_2 (x_c u_c + \sigma) \text{ foot-pounds.} \quad (108)$$

The total work done is

$$W_t = W_1 + W_2 - W_3.$$

Combining above equations and reducing,

$$W_t = (x_1 \rho_1 + q_1 - x_c \rho_c - q_c) \div 778 + p_1 x_1 u_1 - p_c x_c u_c + (p_1 - p_2)\sigma. \quad (109)$$

The last term is small and may be omitted.

Adding and subtracting $\frac{p_c x_c u_c}{778}$ and dividing by 778 equation (109) reduces to

$$H_t = H_1 - H_c + (p_c - p_2)\, x_c u_c \div 778. \quad (110)$$

The steam consumption or water rate of the perfect engine operating in the Rankine cycle with incomplete expansion is

$$W_t' = \frac{2546}{H_1 - H_c + (p_c - p_2) x_c u_c \div 778}, \quad (111)$$

and the efficiency is

$$E_r' = \frac{H_1 - H_c + (p_c - p_2) x_c u_c \div 778}{H_1 - q_2}. \quad (112)$$

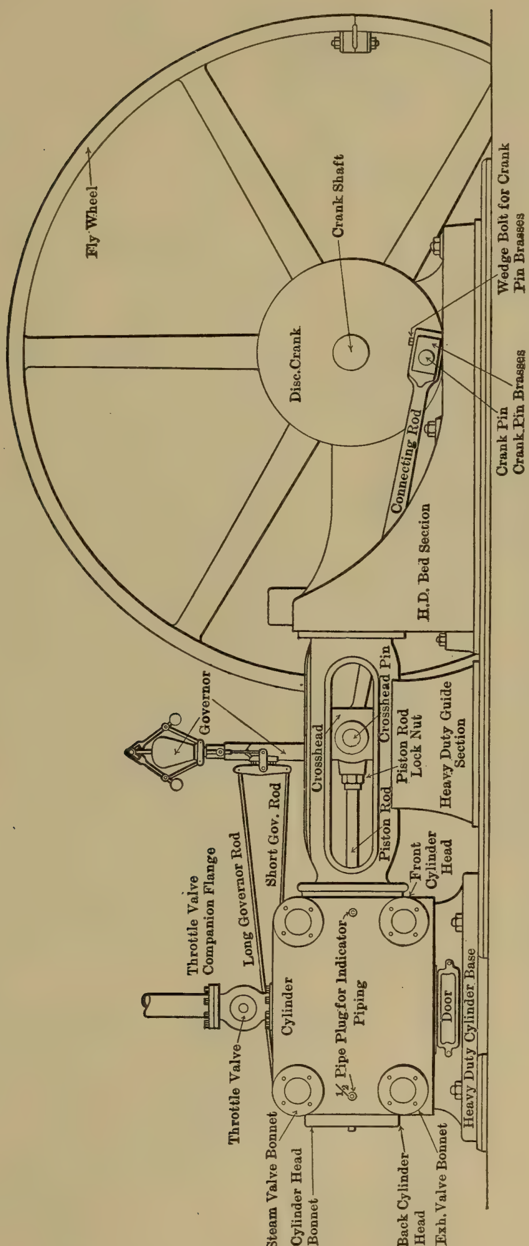


FIG. 188. Side Elevation of a Typical Corliss Engine, Single Cylinder, Belt Drive.

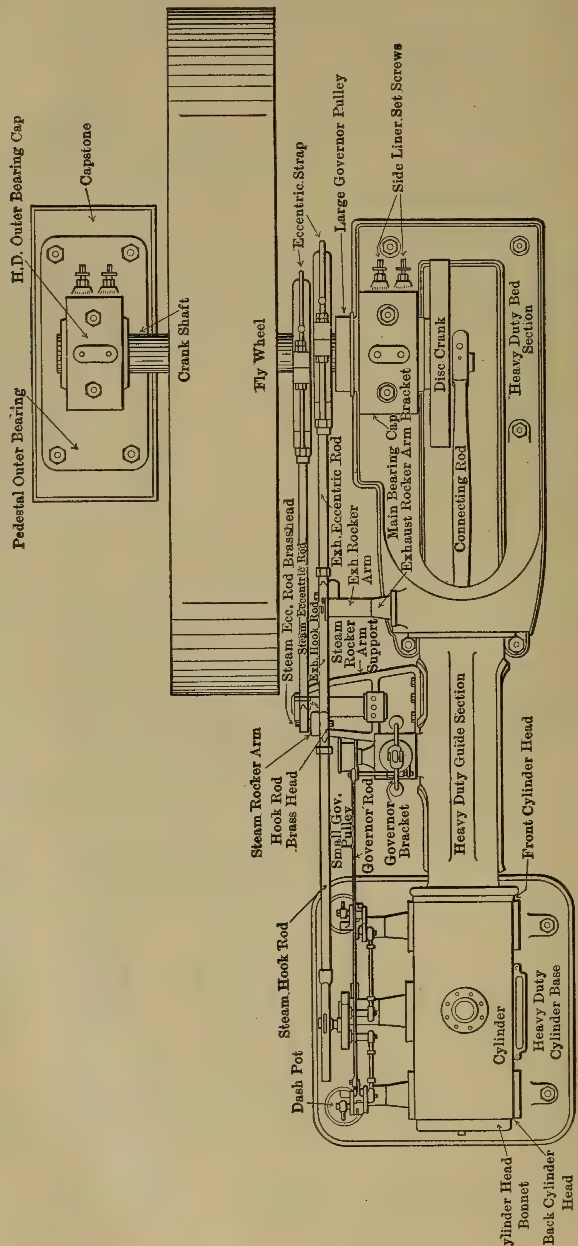


FIG. 189. Plan View of a Typical Corliss Engine, Single Cylinder, Belt Drive.

If p_c is made equal to p_2 equation (112) will be reduced to the same form as equation (99) because the cycle in such case becomes complete.

Example: Find the efficiency and the water rate of a perfect engine working in the incomplete cycle, using the data of the previous example, but assuming release to take place at a pressure of one pound above condenser pressure.

$H_1 = 1332,$ $\quad q_2 = 48$, as previously determined,

$p_c = p_2 + 1 = 0.5 + 1 = 1.5.$

From the Mollier diagram

$H_c = 965,$ $\quad x_c = 0.856.$

From steam tables $u_c = 226$.

Substituting these values in (112)

$$E_r' = \frac{1332 - 965 + (1.5 - 1)\,0.856 \times 226 \div 778}{1332 - 48}$$

$$= 0.286 \text{ or } 28.6 \text{ per cent.}$$

176. Conventional Ideal Engine. — In designing piston engines it is customary to assume as a basis of reference an ideal cycle which considers only the kinetic action of the steam in the cylinder. This permits of analysis without the use of steam tables. The ideal diagram recommended in this connection represents the maximum power obtainable from steam accounted for by the indicator diagram at the point of cut-off.* Such a diagram for a simple non-condensing engine is illustrated in Fig. 190. AB represents admission at pressure p_1, BC represents hyperbolic expansion from cut-off B to release at C and DE represents exhaust at atmospheric pressure p_2. The work done is represented by the area

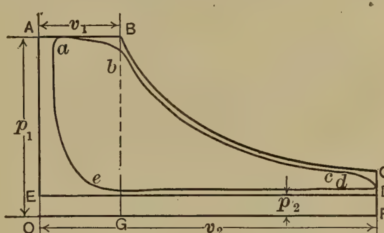


FIG. 190. Conventional Indicator Diagram.

$$ABCDE = OABG + GBCF - OEDF. \qquad (113)$$

Area $\quad OABG = p_1v_1,$

$$GBCF = \int_{v_1}^{v_2} pdv,$$

$$OEDF = p_2v_2.$$

Substituting these values in (113) and reducing

Area $$ABCDE = p_1v_1\left(1 + \log_e \frac{v_2}{v_1}\right) - p_2v_2. \qquad (114)$$

* See Trans. A.S.M.E., vol. 24, p. 751.

Letting $\frac{v_2}{v_1} = r$ (= ratio of expansion),

Area $ABCDE = p_1v_1 (1 + \log_e r) - p_2v_2.$ (115)

The mean effective pressure =

$$\text{M.e.p.} = \frac{\text{area } ABCDE}{v_2}$$

$$= \frac{p_1}{r}(1 + \log_e r) - p_2, \tag{116}$$

The theoretical maximum horse power is

$$\text{H. P.} = \frac{P\,L\,A\,N}{33{,}000}, \tag{117}$$

in which

P = mean effective pressure, pounds per square inch,
L = length of stroke in feet,
A = area of cylinder in square inches,
N = number of working strokes per minute.

The ratio of the m.e.p. of the actual diagram *abcdef* to that of the ideal diagram as determined above is called the *diagram factor.* This factor is determined by experiment and ranges as follows: ("Heat Power Engineering," Hirshfeld and Barnard, 1912, p. 325.)

Simple slide-valve engine	55 to 90
Simple Corliss engine	85 to 90
Compound slide-valve engine	55 to 80
Compound Corliss engine	75 to 85
Triple expansion engine	55 to 70

Example: Determine the probable horse power of a 12-inch × 12-inch simple engine, 250 r.p.m., initial pressure 120 pounds absolute, cut-off $\frac{1}{4}$ stroke, diagram factor, 0.75.

$$r = \frac{v_2}{v_1} = \frac{1}{\frac{1}{4}} = 4.$$

Probable $$\text{m.e.p.} = 0.75 \left\{ \frac{120}{4}(1 + \log_e 4) - 15 \right\}$$

$$= 0.75\,(30 \times 2.386 - 15)$$

$$= 42.4.$$

Probable $$\text{i.h.p} = \frac{42.4 \times 1 \times 113 \times 500}{33{,}000}$$

$$= 72.4.$$

177. The Actual Engine. — To realize the ideal Rankine cycle the walls of the cylinder and the piston must be non-condensing, expansion after cut-off must be adiabatic, the action of the valves must be instantaneous and the steam passages must be sufficiently large to prevent wiredrawing. None of these conditions is fulfilled by the actual engine. The difference between the action of saturated steam in a perfect engine working in the Rankine cycle and that of a simple non-condensing engine for the same initial conditions is shown in Fig. 191 (*A*) and (*B*).

The area *ABCD*, Fig. 191 (*A*), represents the foot-pounds of energy developed per stroke by the ideal engine, and the shaded area *abcd* the energy developed per stroke by the actual engine using the same weight of steam. The difference between the two areas represents the foot-

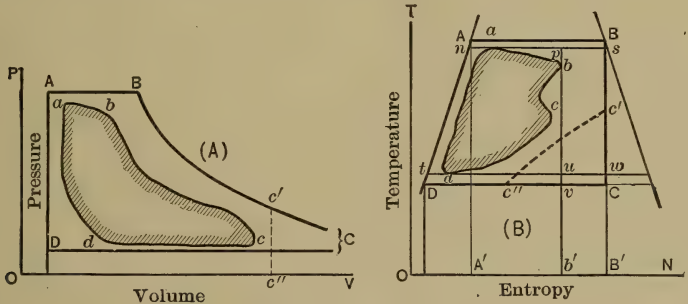


Fig. 191. Imperfections of the Actual Cycle.

pounds of energy lost or wasted. The area *ABCD*, Fig. 191 (*B*), represents the heat available (B.t.u. per stroke) for doing useful work in the ideal engine, and the shaded area *abcd* the heat used by the actual engine. The difference between the two areas represents the heat lost or wasted. The corresponding areas in (*A*) and (*B*) are identical when referred to the same units. The various losses which prevent the actual engine from obtaining the efficiency of the ideal are outlined in paragraphs 187 to 195.

178. Efficiency Standards. — The performance of a steam engine is variously stated as

1. Steam consumption, pounds per hour or per horse-power hour.
2. Heat consumption, B.t.u. per horse power per minute.
3. Thermal efficiency, per cent.
4. Mechanical efficiency, per cent.
5. Efficiency ratio or potential efficiency, per cent.
6. Cylinder efficiency.
7. Commercial efficiency.
8. Duty (see paragraph 309).

Because of the indiscriminate use of many of these terms much confusion arises in comparing the results of different experimenters. The writer has consulted a number of authorities and the definitions given in this text are in accord with the opinion of the majority.

The indicator offers the simplest means of measuring the output of a piston engine, and for this reason the performance is usually stated as *indicated horse power.* The indicated horse power is always greater than the net available power by an amount equivalent to the friction of the mechanism. The power actually developed, or brake horse power, is not readily obtained except for small sizes, and it is customary to approximate this value by deducting the indicated horse power when running idle from the indicated horse power when running under the given load. This does not give the true effective power, but is sufficiently accurate for most commercial purposes. (See paragraph 192.) The output of steam turbines and piston engines driving electrical machinery is conveniently stated in electrical horse power or kilowatts, since the electrical measurements are readily made. The electrical output as measured at the switchboard gives the net effective work, and automatically deducts the machine losses. Large turbines are usually tested at the factory by means of suitable water brakes, and the brake horse power may be obtained from the makers.

179. Steam Consumption. — The most generally used measure of the performance of a steam engine or turbine is the steam consumption per hour or per unit of work output. For reasons stated above the economy of the piston engine is given as the weight of steam consumed per indicated horse-power hour. This must not be confused with the *steam accounted for by the indicator diagram*, or, as it is commonly called, the *indicated steam consumption.* The former refers to the actual weight of fluid flowing through the cylinder and the latter to the weight of steam calculated from the indicator card. (See paragraph 7, Appendix C.) For electrically driven machinery the economy is given as the steam consumption per electrical horse-power hour or per kilowatt hour. If the initial pressure, quality and back pressure were constant for all conditions of operation the hourly steam consumption per unit output would be a true measure of the heat efficiency. (See Tables 59 to 66 for water rates of saturated steam engines and Tables 69 to 73 for superheated steam engines.)

180. Heat Consumption. — Because of the extreme variation in steam conditions the performance of all engines and turbines is best expressed in terms of the heat consumption per unit output measured above the maximum theoretical temperature at which the condensation can be returned to the boiler. This temperature is called the *ideal*

feed-water temperature. Thus the ideal feed-water temperature of an unjacketed non-condensing engine without receiver coils exhausting at standard atmospheric pressure is 212 degrees F., and that of a condensing engine exhausting against an absolute back pressure of two pounds is 126 degrees F. If the engine is fitted with jackets and reheating coils the heat of the liquid at jacket and coil pressure should be added to that of the exhaust in determining the ideal feed-water temperature. For example, if a condensing engine exhausts against an absolute back pressure of two pounds, and ten per cent of the total weight exhausted is condensed in the jackets under a pressure of 150 pounds absolute, the ideal feed-water temperature will be 159.5 degrees F. (Heat of the liquid at 150 pounds absolute = 330 B.t.u. per pound. Heat added by the jackets to the feed water = 330 × 0.1 = 33. Heat of the liquid at two pounds absolute = 94 B.t.u. 94 + 33 = 127 B.t.u., which corresponds to an actual temperature of 159.5 degrees F.)

Example: (1) A compound condensing engine develops one brake-horse-power hour on a steam consumption of 8.5 pounds, initial pressure 200 pounds absolute, superheat 250 degrees F., exhaust pressure 0.5 pound absolute, release pressure two pounds absolute. (2) The same engine when using wet steam develops one brake-horse-power hour on a steam consumption of 12 pounds per hour, initial pressure 150 pounds absolute, quality 98 per cent, exhaust pressure two pounds absolute, release pressure four pounds absolute.

Determine the comparative heat consumption of the two engines:

Superheated steam engine,

$H_1 = 1332$ (from steam tables),

$q_2 = 48$,

Heat supplied per d.h.p.-hour 8.5 (1332 − 48) = 10,914 B.t.u.,

Heat supplied per d.h.p. per minute = 181.9 B.t.u.,

Saturated steam engine,

$$H_1 = x_1r_1 + q_1$$

$$= .98 \times 863.2 + 358.5 = 1176.1,$$

(This may be obtained directly from the Mollier diagram.)

$q_2 = 94$,

Heat supplied per d.h.p.-hour = 12 (1176 − 94) = 12,985 B.t.u.,

Heat supplied per d.h.p. per minute = 216.4.

Economy of superheated steam

(1) in steam consumption, $100\dfrac{12 - 8.5}{12} = 29.2$ per cent,

(2) in heat consumption, $100\dfrac{216.4 - 181.9}{216.4} = 15.9$ per cent.

181. Thermal Efficiency. — The *thermal efficiency* of a steam engine or turbine is the ratio of the heat converted into *useful* work to that *supplied,* measured above the heat of the liquid at exhaust pressure.* If the heat consumption is expressed in terms of i.h.p.-hour, the ratio becomes the *indicated thermal efficiency.* Since the heat equivalent of

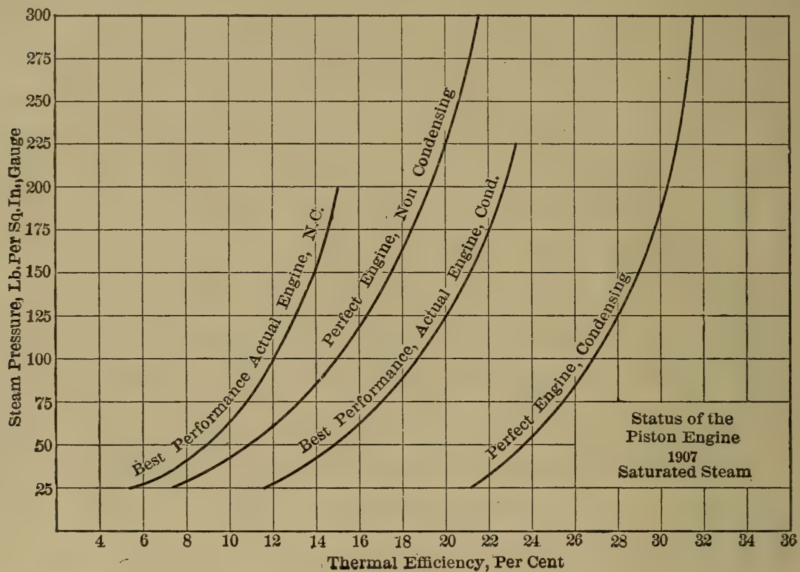


Fig. 192.

one horse power, using the latest accepted values, is 42.44 B.t.u. per minute or 2546 B.t.u. per hour, this relationship may be expressed

$$E_t = \frac{2546}{\text{B.t.u. supplied per b.h.p.-hour}}$$

$$= \frac{2546}{W(H_1 - q_2)}, \tag{118}$$

in which

W = the weight of steam supplied, pounds per developed horsepower hour,

H_1 and q_2 as in previous equations.

* The heat supplied is often measured above the *actual* feed-water temperature but the latter is not dependent upon the performance of the engine and hence is not satisfactory for purposes of comparison. The ratio of the heat converted into useful work to that supplied above the *actual* feed-water temperature is designated as the "thermal efficiency ratio," in the A.S.M.E. Code for Testing Steam Engines.

If measured in electrical units this relationship becomes

$$E_t' = \frac{3413}{W_1 (H_1 - q_2)}, \tag{119}$$

in which

W_1 = pounds per kilowatt-hour, other notations as in (118).

Example: Determine the thermal efficiency for the two engines using the data of the preceding problem.

Superheated steam engine,

$$E_t = \frac{2546}{8.5 (1332 - 48)} = 0.233 = 23.3 \text{ per cent.}$$

Saturated steam engine,

$$E_t = \frac{2546}{12 (1176.1 - 94)} = 0.196 = 19.6 \text{ per cent.}$$

Fig. 192 shows the present status of the piston engine, using saturated steam for various initial pressures. Table 66 gives the thermal efficiencies for saturated steam engines, and Table 69 the thermal efficiencies for superheated steam engines.

182. Mechanical Efficiency. — The ratio of the developed or brake horse power to the indicated power is the mechanical efficiency of the engine; the ratio of the electrical horse power to the indicated power is the mechanical efficiency of the engine and generator combined; and the ratio of the pump horse power to the indicated power of the engine is the mechanical efficiency of the engine and pumps combined. The percentage of work lost in friction is therefore the difference between 100 per cent and the mechanical efficiency in per cent. (See also paragraph 192.) Table 56 gives the mechanical efficiency for several types of engines and Fig. 193 illustrates the relation between load and mechanical efficiency for a 75-kilowatt direct-connected engine and generator.

183. Efficiency Ratio. — The degree of perfection of an engine or the extent to which the theoretical possibilities are realized is the ratio of the thermal efficiency of the actual engine to that of an ideally perfect engine working in the Rankine cycle with complete expansion. This is called the *efficiency ratio* or *potential efficiency*.* It is the accepted standard for comparing the performance of steam engines and steam turbines.

* The term "thermodynamic efficiency" or "efficiency" without qualification is ordinarily interpreted as the efficiency ratio, though some authorities apply the name "thermodynamic efficiency" to the "thermal efficiency" as defined in paragraph 181.

TABLE 56.

MECHANICAL EFFICIENCIES OF ENGINES.

Kind of Engine.	Horse Power.	Efficiency at Full Load.
Simple:		
1. High-speed, non-condensing	150	95.5
2. High-speed, condensing	170	96
3. Low-speed, non-condensing	275	94
Compound:		
4. High-speed, non-condensing	150	94
5. High-speed, condensing	160	98
6. Low-speed, non-condensing	900	95
7. Low-speed, condensing	1000	95
8. Do	5500	95.2*
9. Do	7500	93.0*
Triple: (combined efficiency of engine and pump)		
10. Pumping engine	865	97.4
Quadruple: (combined efficiency of engine and pump)		
11. Pumping engine	712	93

* Combined efficiency of engine and generator.

1. Buffalo Simple engine, 12 × 12, Elec. World, Sept., 1904, p. 147.
2. Reeves Simple engine, 15 × 14, Elec. World, Oct. 1, 1904, p. 587.
3. 24 × 48 Hamilton Corliss at Armour Inst. of Tech., 1898.
4. Reeves Compound, Eng. Rec., July 1, 1905, p. 24.
5. Reeves Compound, Eng. Rec.
6. 21, 41 × 30 Cross Compound Ball & Wood, West Albany Station, N.Y.C. & H.R.R.
7. 20, 40 × 42 Rice and Sargent, A.S.M.E., 29–1276.
8. N.Y. Edison, Waterside Station.
9. New York Subway.
10. Allis Pumping Engine, Power, May, 1906, p. 299.
11. Nordburg Pumping Engine, Eng. News, May 4, 1899, p. 280.

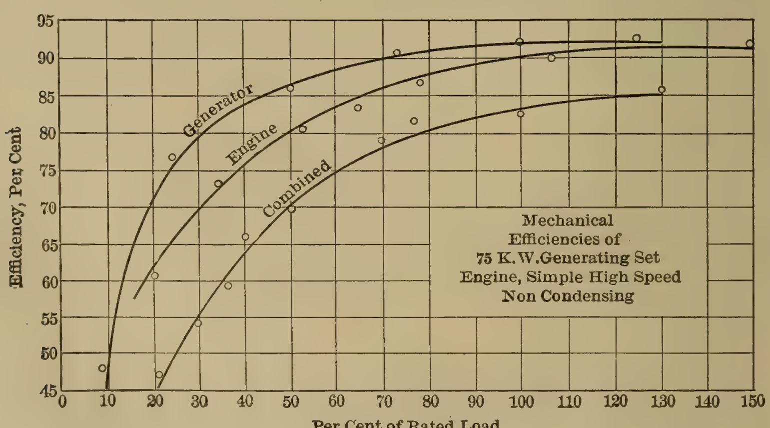


FIG. 193.

If E = efficiency ratio,
E_t = thermal efficiency of the actual engine,
E_r = efficiency of the ideal engine work in the Rankine cycle with complete expansion,

Then
$$E = \frac{E_t}{E_r}. \tag{120}$$

From equation (118),
$$E_t = \frac{2546}{W_1(H_1 - q_2)}.$$

And from equation (99)
$$E_r = \frac{H_1 - H_2}{(H_1 - q_2)}.$$

Whence
$$E = \frac{2546}{W_1(H_1 - q_2)} \div \frac{H_1 - H_2}{H_1 - q_2} \tag{121}$$
$$= \frac{2546}{W\,(H_1 - H_2)}. \tag{122}$$

Example: Determine the efficiency ratio of the two engines specified in paragraph 180.

Superheated steam engine:
$$E = \frac{2546}{8.5\,(1332 - 908)} = 0.706 = 70.6 \text{ per cent.}$$

Saturated steam engine:
$$E = \frac{2546}{12\,(1176 - 898)} = 0.763 = 76.3 \text{ per cent.}$$

Tables 66 and 69 give the best recorded efficiency ratios for current practice.

184. Cylinder Efficiency. — The piston engine seldom expands the steam down to the existing back pressure but releases from two to five pounds above this point in condensing engines and from 15 to 20 pounds above in non-condensing engines. The ideal cycle corresponding to this condition is the Rankine cycle with incomplete expansion. The ratio of the thermal efficiency of the actual engine to that of the ideal engine working in the incomplete cycle is a true measure of the degree of perfection of the engine under the given conditions. This rate is called *cylinder efficiency* and may be expressed as

$$E' = \frac{2546}{W[(H_1 - H_c) + (p_c - p_2)\,x_c u_c \div 778]}. \tag{123}$$

See equations (118) and (112).

It may be expressed also by the relationship

$$E' = \frac{\text{Steam consumption of the ideal engine with incomplete expansion}}{\text{Steam consumption of the actual engine}}.$$

Example: Determine the cylinder efficiency of the two engines specified in paragraph 180.

Superheated steam engine,

$$E' = \frac{2546}{8.5^* [1332 - 980 + (2.0 - 0.5)\, 0.866 \times 173.5 \div 778]}$$

$$= 0.85 = 85.5 \text{ per cent.}$$

Saturated steam engine,

$$E' = \frac{2546}{12\, [1176 - 935 + (4 - 2)\, 0.808 \times 90.5 \div 778]}$$

$$= 0.88 = 88 \text{ per cent.}$$

Summing up the various efficiencies for the two cases analyzed in paragraphs 180 to 184:

	Saturated Steam Engine.	Superheated. Steam Engine.
Pressure, pounds per square inch, absolute:		
Initial	150	200
Release	4	2
Condenser	2	0.5
Degree of superheat, degrees F.	0.98*	250
Steam consumption, pounds per developed-horse-power hour		
Actual engine	12.00	8.50
Ideal engine, Rankine cycle, with incomplete expansion	10.56	7.22
Ideal engine, Rankine cycle, with complete expansion	9.16	6.00
Ideal engine, Carnot cycle	8.30	....
Thermal efficiency, per cent		
Actual engine	19.6	23.3
Ideal engine, Rankine cycle, with incomplete expansion	22.3	27.4
Ideal engine, Rankine cycle, with complete expansion	25.8	33.3
Ideal engine, Carnot cycle	28.3	....
Heat consumption, B.t.u., per developed-horse-power minute	216.4	181.9
Actual engine, Ideal engine, Rankine cycle, with incomplete expansion	190.4	154.6
Ideal engine, Rankine cycle, with complete expansion	165.1	128.5
Ideal engine, Carnot cycle	149.7	......
Efficiency ratio, per cent	76.3	70.6
Cylinder efficiency, per cent	88.0	85.0

* Quality.

* If the steam consumption per i.h.p.-hour is used in this connection instead of the consumption per d.h.p.-hour this ratio becomes the *indicated* cylinder efficiency.

185. Commercial Efficiencies. — There is no accepted standard for rating the commercial efficiency of an engine or turbine. The various measures used in this connection, such as *pounds* of *standard coal per d.h.p.-hour, cents per horse power per year* and the like include the economy of the boiler and auxiliaries and are not a true indication of the performance of the engine alone. From a commercial standpoint it is important to know the weight of coal required to develop a horse-power hour, taking into consideration all of the losses of transmission and conversion, and a knowledge of the *over-all efficiency* from switchboard to coal pile is of value in basing the cost of power, but these items are in reality measures of the plant economy and are of little value in comparing the performance of the prime mover. The various efficiencies under this heading are treated in Appendices C to G.

186. Heat Losses in the Steam Engine. — The principal losses which tend to lower the efficiency of the steam engine and which prevent it from realizing the performance of the ideal engine are due to

(*a*) Cylinder condensation.
(*b*) Leakage.
(*c*) Clearance volume.
(*d*) Incomplete expansion.
(*e*) Wire drawing.
(*f*) Friction of the mechanism.
(*g*) Presence of moisture in the steam at admission.
(*h*) Radiation, convection and minor losses.

187. Cylinder Condensation. — The weight of steam apparently used per revolution, as determined from the indicator card, or the *indicated steam consumption*,* (see paragraph 7, Appendix C) is considerably less than that actually supplied. The difference or *missing quantity* is due chiefly to *cylinder condensation.* This is by far the greatest loss in the steam engine with the exception of that inherent in the ideal engine. When steam is admitted to the cylinder a considerable portion of the heat is given up to the comparatively cool skin surface of the cylinder walls. If the steam is saturated at admission this heat absorption causes condensation, or *initial condensation* as it is called; if superheated at admission the temperature is lowered to a corresponding point. After cut-off heat continues to be given up to the walls until the temperature of the steam falls below that of the skin surface, when the process is reversed and part of the heat is returned to the steam. With saturated steam the heat absorption causes *condensation during expansion* and the heat rejected, *reëvaporation during expansion.*

* Also called the *steam accounted for by the diagram* or *diagram steam.*

With superheated steam an equivalent heat exchange takes place. Unless the cylinder is of a compound series the heat absorbed from the cylinder walls during exhaust does no useful work and is lost. Cylinder condensation, measured as the proportion of the mixture present, varies with the size of the engine, speed, length of cut-off, valve design, temperature range, location of ports and port passages, jacketing, lagging, and other variables. It ranges from 16 to 30 per cent, and is occasionally as high as 50 per cent of the total weight of steam admitted to the cylinder. Cylinder condensation and leakage are

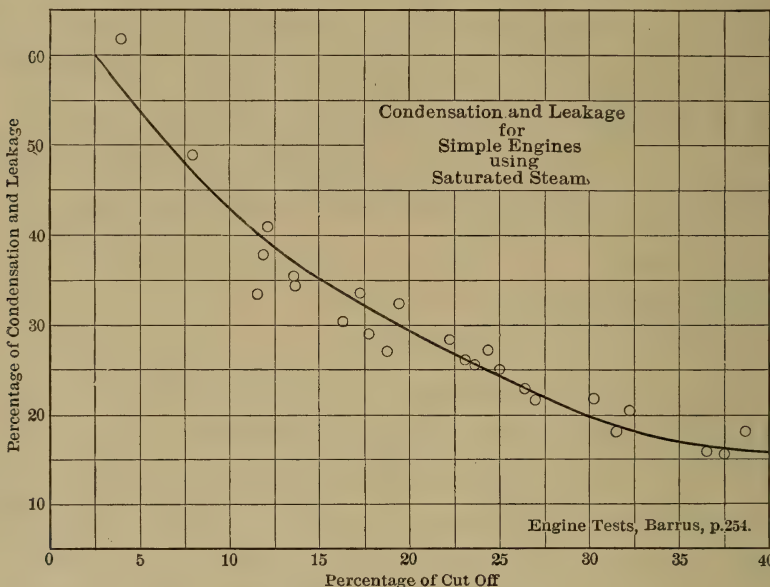


FIG. 194.

ordinarily classified together for sake of simplicity. Fig. 194 shows the relation between cylinder condensation and leakage losses for various percentages of cut-off for simple high-speed non-condensing engines.

Empirical formulas for calculating the extent of these losses, and which involve the various influencing factors, are unwieldly and only approximately accurate. One of the most satisfactory formulas of this class is that deduced by R. C. H. Heck, "The Steam Engine and Turbine," 1911, p. 175.

The various heat exchanges between working fluid and the cylinder walls, including cylinder condensation and leakage, are approximately determined by transferring the indicator diagram to the temperature entropy chart. (See Fig. 191.)

For use and application of the temperature-entropy diagram in engine tests consult Power, Dec., 1907, p. 834; Jan. 21, 1908, p. 96; Jan. 28, 1908, p. 145.

A comparatively simple method for approximating cylinder condensation and leakage losses is given by J. Paul Clayton, Proc. A.S.M.E., April, 1912, and consists in transferring the indicator diagram to logarithmic cross-section paper. By means of the logarithmic diagram Clayton found that (1) free from certain abnormal influences, expansion and compression take place in the cylinder substantially according to the law $PV^n = C$, (2) the value n bears a definite relation in any given cylinder to the proportion of the total weight of steam mixture which was present as steam at cut-off, (3) the relation of the value n to the value X_c (quality of steam at cut-off) for the same class of cylinder as

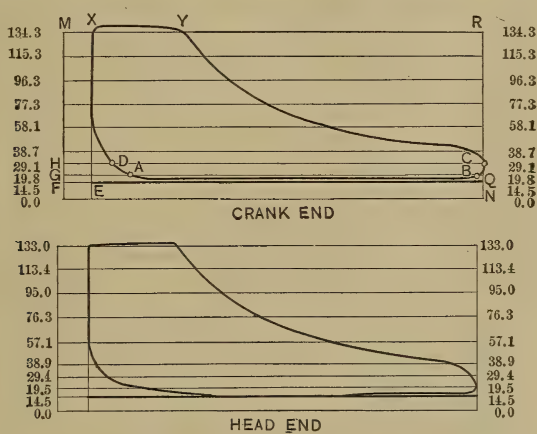


FIG. 195. Diagrams taken from a 12 × 24 Corliss Engine.

regards jacketing is practically independent of engine speed and of cylinder size, and (4) by means of the experimentally determined relation of X_n and n the actual steam consumption may be obtained from the indicator card to well within 4 per cent of the true value. The curves in Fig. 196 were plotted on logarithmic cross-section paper from the pressure-volume diagrams, shown in Fig. 195, and illustrate Mr. Clayton's method of analysis. The curves in Figs. 197 and 198 show the relation between quality X_c and exponent n or for a given set of conditions. For applications of this method to concrete examples with a full discussion of the results consult the paper referred to.

188. Leakage of Steam. — The loss due to leakage is a variable factor depending upon the design and condition of the engine, and is greater with saturated than with superheated steam. The usual

method of measuring leakage past the valves and piston while the engine is at rest is likely to give erroneous results, as demonstrated by Callender and Nicolson (Peabody, "Thermodynamics," p. 351) in tests

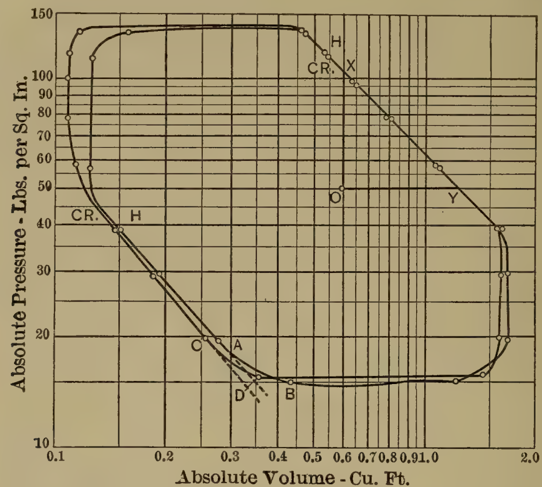


Fig. 196. Logarithmic Diagrams plotted from Fig. 195.

made on a high-speed automatic balanced valve engine and on a quadruple expansion engine with plain unbalanced slide valves. With the engines at rest they found that the leakage past valves and piston was

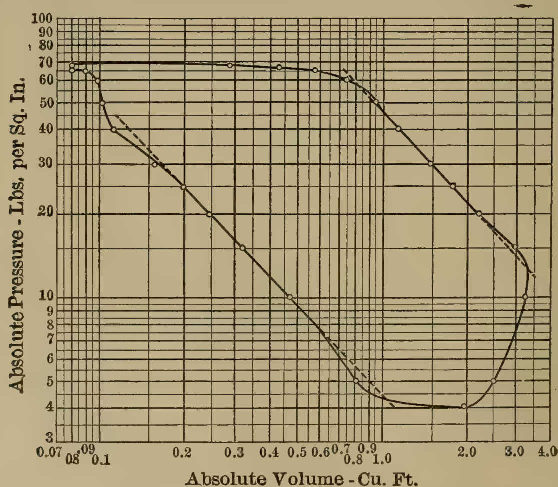


Fig. 197. Diagram from a 14 × 35 Corliss Engine, showing Leakage at Beginning and End of Expansion and Compression.

insignificant, but when in operation the leakage from the steam chest into the exhaust was very considerable indeed. It was thought that a

large proportion of the leakage was probably in the form of water formed by condensation of steam on the seat uncovered by the valve.

According to the report of the Steam Engine Research Committee (Eng. Lond., March 24, 1905, p. 208), leakage through a plain slide valve is independent of the speed of the sliding surfaces, and directly proportional to the difference in pressure on the two sides; with well-fitted valves the leakage is never less than 4 per cent of the volume of steam entering the cylinders, and is often greater than 20 per cent.

The various leakage losses may be approximated by transferring the indicator diagram to logarithmic cross-section paper. Fig. 197 shows the application of the logarithmic diagram to a specific case and illustrates this method of determining leakage losses. See Clayton's paper referred to in paragraph 188.

Leakage Past Piston Valves: Engr., Feb. 9, 1912.

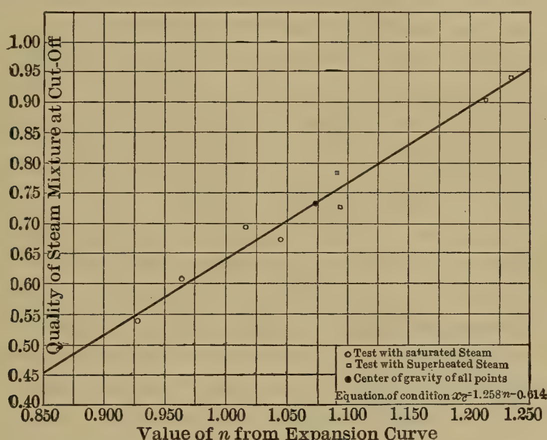


FIG. 198. Relation of Quality and the Value of n.

189. Clearance Volume. — The portion of the cylinder volume not swept through by the piston but which is nevertheless filled with steam when admission occurs is called the clearance volume. It is the space between the end of the piston when on dead center and the inside of the valves covering the ports. It varies from about 1 per cent of the piston displacement in very large engines with short steam passages to 10 per cent or more in small high-speed engines. When the steam retained in the clearance space is compressed to the initial pressure and expansion is carried down to the back pressure, the clearance has little effect upon the economy of the engine, but since expansion and compression are seldom complete in actual practice, the loss may be considerable. (Ripper, "Steam Engine," p. 103.) The shorter the cut-off

the greater will be the ratio of the weight of cushion steam to that of the steam supplied and hence the greater the relative loss. In large slow-speed engines the loss may be insignificant if the clearance volumes are small, while in small high-speed engines it may be considerable.

The ratio of expansion is decreased by clearance; for example, an engine cutting off at one-fifth, neglecting clearance, has an apparent ratio of expansion of 5, but if the clearance volume is 10 per cent the actual ratio is only 3.66. One of the few recorded tests relative to the influence of clearance on the economy of a high-speed engine was conducted on a 14 × 15 Allfree engine. (Power, May, 1901.) With a clearance volume of 2.2 per cent, initial pressure 105 pounds gauge, and 172 r.p.m., the best performance was 23.7 pounds of dry steam per i.h.p. hour. With the same steam pressure and speed, but with clearance volume increased to 6 per cent by the use of a shorter piston, the best performance was 28.3 pounds per i.h.p. hour. In both cases the compression was carried up to admission pressure.

Independent tests made by Prof. Boulvin and by A. H. Klemperer on single-cylinder Corliss engines gave a minimum water rate when the clearance volume was approximately one-half the compression volume. See end of paragraph 190.

Engine Clearance and Compression: Power, July 5, 1910, Dec. 27, 1910; Sibley Journal, Dec., 1910.

190. Loss due to Incomplete Expansion and Compression. — Theoretically the loss due to incomplete expansion is considerable. For example, the theoretical steam consumption of a perfect engine (Rankine cycle) expanding from 120 pounds absolute to a condenser pressure of 2 pounds absolute is 9.6 pounds per horse-power hour. If the expansion were carried to only 5 pounds absolute, the exhaust pressure remaining the same, the steam consumption would be increased to 11.8 pounds per horse-power hour, a difference of 22 per cent for an increase in terminal pressure of only 3 pounds per square inch. The theoretical water rates for various terminal pressures are given below.

Terminal Pressure, Pounds per Square Inch Absolute.	Steam Consumption of Perfect Engine.	Terminal Pressure, Pounds per Square Inch Absolute.	Steam Consumption of Perfect Engine.
1	8.5	3	10.4
1.5	9.1	4	11.1
2	9.6	5	11.8
2.5	10	6	12.3

In actual engines expansion is seldom complete, since it would necessitate increased bulk and weight of engine, and the work done

by the steam in the last stages would not compensate for the increased cost.

In single-cylinder engines maximum economy is effected when the terminal pressure is considerably above that of the exhaust, since the gain due to complete expansion is more than offset by the increased cylinder condensation. This is true to a certain extent in all engines irrespective of the number of cylinders. Tests by G. H. Barrus ("Engine Tests," 1900) to determine the terminal pressures effecting maximum economy for various types of engine gave results as follows:

	Terminal Pressure, Pounds Absolute.
Simple slide-valve engines, non-condensing	30 to 40
Simple slide-valve engines, condensing	25 to 30
Simple Corliss engines, non-condensing	20 to 25
Simple Corliss engines, condensing	15 to 18
Compound engines, non-condensing	18 to 22
Compound engines, condensing	3 to 5

In high-speed engines a certain amount of compression is desirable for its cushioning effect; outside of this mechanical feature compression may or may not be of benefit to the engine, as will be seen from the results of tests stated below. Zeuner in his treatise on theoretical thermodynamics proves deductively that in an engine with a large clearance volume the loss due to clearance is completely eliminated if the compression is carried up to admission pressure, a conclusion which tests by Jacobus, Carpenter, and others fail to confirm. A series of tests by Professor Jacobus (Trans. A.S.M.E., 15–918) on a 10 × 11 high-speed automatic engine at Stevens Institute show decreasing economy with increase of compression, the initial pressure, cut-off, and release remaining constant. The results were as follows:

Proportion of initial pressure up to which the steam is compressed	$\frac{5}{8}$	$\frac{2}{3}$	Full
Steam, pounds per i.h.p. hour	34.8	36.7	38

Tests by Carpenter (Trans. A.S.M.E., 16–957) on the high-pressure cylinders of the Corliss engine at Sibley College gave:

Compression, per cent	11.4	25	35.2
Brake horse power	30	29	26
Steam, pounds per b.h.p. hour	33	33.3	34

Tests, made by A. H. Klemperer on a 7.1 × 17.7-inch Corliss engine, at Dresden, gave decreasing steam consumption for increase in com-

pression up to a compression of about twice the clearance volume, beyond which the water rate increased with the increase in compression. (Zeit d. Ver. deut. Ingr., Vol. I, 1905, p. 797.)

Tests, made by Prof. Boulvin on a 9.8 × 19.7-inch Corliss engine at University of Ghent, give results agreeing with those of Klemperer. (Revue de Mécanique, 1907, Vol. XX, p. 109.)

Fig. 199 shows the influence of increasing back pressure on the economy of an 8 × 10-inch automatic high-speed engine at the Armour Institute of Technology.

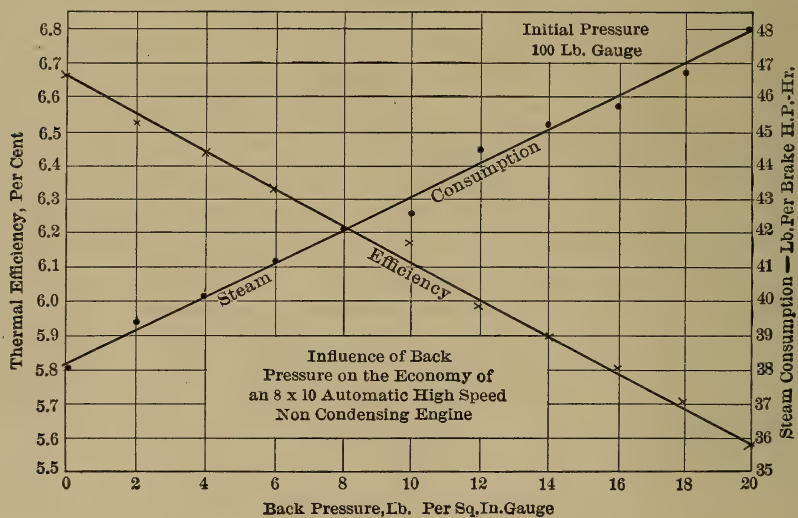


Fig. 199.

191. Loss due to Wire Drawing.— Wire drawing, or the drop in pressure due to the resistances of the ports and passages, has the effect of reducing the output and the economy of the engine to some extent, since the pressure within the cylinder is less than that at the throttle during admission and greater than discharge pressure at exhaust. The steam may be dried to a small extent during admission, but because of the drop in pressure the *heat availability* is reduced. In single-valve engines the effects of wire drawing are decidedly marked and the true points of cut-off and release are sometimes difficult to locate on the indicator card. In engines of the Corliss or gridiron-valve type the effects are hardly noticeable.

192. Loss due to Friction of the Mechanism. — The difference between the indicated horse power and that actually developed is the power consumed in overcoming friction, and varies from 4 to 20 per cent of the indicated power, depending upon the type and condition of the

engine. Engine friction may be divided into (1) initial or no-load friction and (2) load friction. The stuffing-box and piston-ring friction is practically independent of the load, while that of the guides, bearings, and the like increases with the load. In Fig. 200, curve *A* gives the relation between the frictions for a four-slide-valve horizontal cross compound engine, and *B* that for a simple non-condensing Corliss. (Peabody's "Thermodynamics," pp. 433 and 437.) Curve *C* is plotted

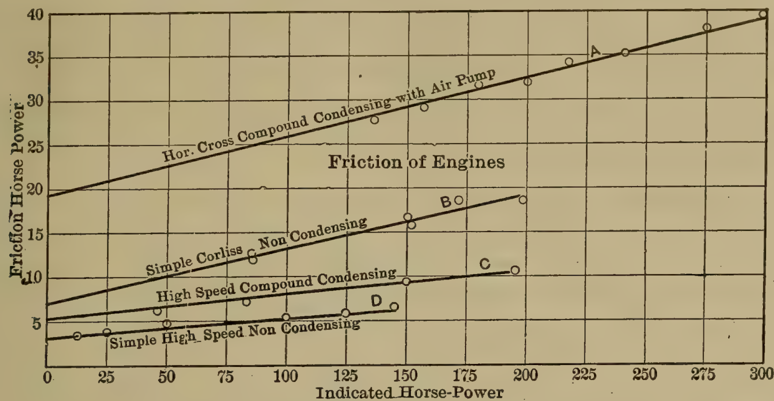


FIG. 200. Typical Curves of Steam Engine Friction.

TABLE 57.

DISTRIBUTION OF FRICTION IN SOME DIRECT-ACTING STEAM ENGINES.

(Thurston.)*

Parts of Engines where Friction is Measured.	Percentage of Total Engine Friction.				
	"Straight Line" Balanced Valve.	"Straight Line" Unbalanced Valve.	Traction Engine Locomotive Valve Gear.	Automatic Balanced Valve.	Condensing Engine Balanced Valve.
Main bearings..................	47.0	35.4	35.0	41.6	46.0
Piston and piston rod.........	32.9	25.0	21.0	49.1	21.8
Crank pin....................	6.8	5.1	13.0		
Crosshead and wrist pin.......	5.4	4.1			
Valve and valve rod..........	2.5	26.4	22.0	9.3	21.0
Eccentric strap...............	5.4	4.0			
Link and eccentric............	....	....	9.0	....	....
Air pump....................	....	....	....	....	12.0
	100.0	100.0	100.0	100.0	100.0

* "Friction and Lost Work in Machinery," p. 13.

from the tests of a Reeves vertical cross compound condensing engine (Engineering Record, July 1, 1905, p. 24), and D from the test of an Ames simple high-speed non-condensing engine. (Engineering Record, Vol. 27, p. 225.) A large number of recorded tests show less friction at full load than at no load, but this is probably due to error or to variations in lubrication. With first-class lubrication it is usually sufficiently accurate to assume the friction to be constant and equal to the initial friction at zero load. The distribution of the frictional losses in a number of engines is given in Table 57.

193. Moisture. — The presence of moisture in the steam pipe is due to condensation caused by radiation or to priming at the boiler. Unless removed by some separating device between boiler and engine the amount of moisture entering the cylinder may be from 1 to 5 per cent of the total weight of steam, and the work done per pound of fluid is correspondingly reduced. This loss should not be charged against the engine, however, and its performance should be reckoned on the dry steam basis. Experiments reported by Professor R. C. Carpenter (Trans. A.S.M.E., 15–438) in which water in varying quantities was introduced into the steam pipe, causing the quality of the steam to range from 99 per cent to 57 per cent, showed that the consumption of *dry steam* per i.h.p. hour was practically constant, the water acting as an inert quantity. An efficient separator will remove practically all the entrained water.

194. Radiation and Minor Losses. — The radiation and conduction of heat from the cylinder, piston rod and valve stem has the effect of increasing the cylinder condensation. In jacket engines this loss may be approximated by the quantity of steam condensed in the jacket when the engine is not running. In unjacketed engines the loss is practically underterminable since the heat exchange between cylinder walls and the steam is exceedingly complex.

195. Heat Lost in the Exhaust. — Most of the heat supplied to the engine is rejected to the exhaust; this varies from 65 per cent in the best types of engines to 95 per cent in the poorer types. If all of the exhaust is used for heating or manufacturing purposes the heat chargeable to power is the difference between the heat supplied and that rejected. In determining the latter it is necessary to know the quality of the exhaust steam since a considerable portion of the fluid discharged is water. The quality varies from 92 per cent in high-speed non-condensing engines running at full load to 80 per cent or lower in compound non-condensing engines operating at light loads. For example, a 24 × 18-inch simple engine direct connected to a 200-kilowatt generator, installed at the Armour Institute of Technology, uses 55 pounds

of steam per kilowatt hour at full load, initial pressure 115 pounds absolute, back pressure 2 pounds gauge. The quality of the exhaust entering the heating system is 90 per cent. During the summer months when the exhaust is discharged to waste the entire heat supplied above the feed-water temperature of 210 degrees F. is chargeable to power, an extravagant waste of heat. During the winter months when all of the exhaust is used for heating purposes the heat chargeable to power is $55 [1188.8 - (0.9 \times 965.6 + 187.5)] = 7276$ B.t u. per kilowatt-hour, which is equivalent to $7276 \div [1188 - (210 - 32)] = 7.2$ pounds of boiler steam per kilowatt-hour, a performance unequalled by any compound condensing engine. With condensing engines, in which no disposition is made of the heat absorbed by the circulating water, which is the usual case, all of the heat rejected to the exhaust less the small amount reclaimed from the hot well is chargeable to power, The latent heat rejected to exhaust, however, is an inherent loss, even for the ideal engine operating in the Rankine cycle.

196. Methods for Increasing Economy. — Various methods have been adopted for bettering the economy of piston engines, among them may be mentioned:

(*a*) Increasing boiler pressure.
(*b*) Use of receiver reheaters.
(*c*) Steam jackets.
(*d*) Increasing rotative speed.
(*e*) Compounding.
(*f*) Superheating.
(*g*) Decreasing back pressure.
(*h*) Use of binary vapors.
(*i*) Use of uniflow or straight flow cylinders.

Some of these items will be considered separately, others will be included in the discussion of the different classes of engines.

197. Effect of Increased Steam Pressure. — A consideration of the Rankine and Carnot Cycles indicates that theoretically the greater the temperature range the greater will be the efficiency. (See Fig. 192.) In the actual engine the temperature range is most readily increased by raising the boiler pressure, since the limit of the back pressure is practically fixed by the cooling medium in the condenser. The theoretical gain resulting from increased pressure range is, however, very considerably affected by the increased losses due to cylinder condensation.

Fig. 201 shows the results of tests made at the Armour Institute of Technology on an 8×10 automatic high-speed piston-valve engine, showing marked gain with increase of initial pressure up to a certain

point when the condensation losses became sufficiently great to neutralize the advantage which would otherwise be gained.

The following figures were obtained in tests of a small Willans engine, non-condensing, under different steam pressures:

Initial Pressure, Gauge.	Pounds Steam per I.H.P. Hour.	B.T.U. per I.H.P. per Minute.
36.3	42.8	700
51.0	36.0	595
74.0	32.6	544
85.0	29.7	495
97.0	26.9	450
110.0	27.8	465
122.0	26.0	436

Referring to Fig. 192, it may be noted that both the theoretical and the actual efficiencies increase very slowly for pressures above 150 pounds. Practically, gain in efficiency due to increasing the pressure

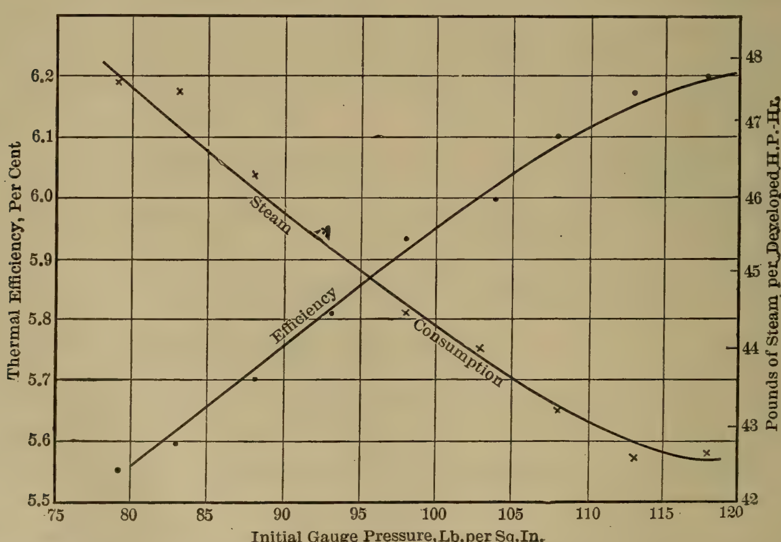


FIG. 201. Influence of Initial Pressure on the Economy of a Small, High-speed, Non-condensing Engine.

above about 200 pounds is at the expense of increased first cost and maintenance and is only resorted to when small weight and space are the most important considerations.

The range of pressures sanctioned by modern practice for different types of engines is as follows:

Type of Engine.	Range in Pressure (Gauge).	Average.
Simple slow-speed (standard type)	60–120	90
Simple high-speed (standard type)	70–125	100
Simple, uniflow	125–225	175
Compound high-speed, non-condensing	100–180	140
Compound high-speed, condensing	100–180	140
Compound slow-speed, condensing	125–200	170
Triple expansion, condensing	140–250	200
Quadruple expansion, condensing	125–275	225

198. Receiver reheaters: Intermediate Reheating.—The receivers between the cylinders of multi-expansion engines are frequently equipped with heating coils, as illustrated in Fig. 447, the function of which is to superheat the exhaust steam before delivering it to the cylinder immediately following, with a view of reducing the losses occasioned by cylinder condensation. The coils are supplied with live steam under boiler pressure and may serve to evaporate a portion of the moisture or to actually superheat the steam supplied to the following cylinder. The question of the propriety of using reheaters is an open one, since reliable data relative to their use are meager and discordant. The conditions under which the few recorded tests were made are too diverse to warrant definite conclusions. Some show an appreciable gain in economy, others a decided loss. A reheater is of little value in improving the thermodynamic action of the engine, and is probably a loss unless it produces a superheat of at least 30 degrees F., and to be fully effective should superheat above 100 degrees F. (L. S. Marks, Trans. A.S.M.E., 25–500.) The effectiveness of the reheater will evidently be increased by the removal of the greater portion of the moisture from the exhaust steam before it enters the receiver. In the 5000-horsepower engine at the Waterside Station in New York it was shown that both jackets and reheaters, either together or alone, were practically valueless throughout the working range of load. (Power, July, 1904, p. 424.) Many similar cases may be cited which show no gain in economy with the use of the reheaters. In all cases the reheater effects a great reduction in the condensation in the low-pressure cylinders, but the resulting gain, considering the condensation in the reheater coils, may be little if any. On the other hand, with properly proportioned reheaters, the gain may be considerable and particularly with superheated steam. Practically all European engines operating with highly superheated steam are equipped with receiver-reheaters. In

the locomobile type of engine plant the intermediate reheating is effected by heating coils placed in the path of the furnace gases. For a complete description of the locomobile with results of tests under varying conditions of operations, see Zeit. des. Ver. deut. Ingr., Vol. 55, 1911, p. 410 (serial); ibid, p. 922. See also Fig. 226.

In triple-expansion pumping engines receiver-reheaters are found to effect an appreciable gain in economy, and practically all such engines are equipped with them. In electric traction work or where the load is a widely fluctuating one the reheater has been virtually abandoned. Apart from the consideration of fuel economy, all tests show a marked increase in the indicated power of the low-pressure cylinder (5 to 15 per cent), and to that extent it increases the capacity of the entire engine. (G. H. Barrus, Power, Sept., 1903, p. 516.)

Engine Reheaters: Mech. Engr., Dec. 23, 1910.

199. Jackets. — If the walls of the cylinder are made double and the space between is filled with live steam under boiler pressure, the cylinder is said to be steam jacketed. The function of the jacket is to reduce initial condensation by maintaining the temperature of the internal walls as nearly as possible equal to that of the entering steam. The heat given up by the jacket steam, and the resulting condensation, is usually a smaller loss than would otherwise result from cylinder condensation. However, tests of numerous engines with and without steam jackets do not agree as to the conditions under which their use is profitable, the apparent gain ranging from zero to 30 per cent. According to Peabody, a saving of from 5 to 10 per cent may be made by jacketing simple and compound condensing engines, and a saving of from 10 to 15 per cent by jacketing triple expansion engines of 300 horse power and under. On large engines of 1000 horse power or more the gain, if any, is very small. (Peabody, "Thermodynamics," p. 400.)

Other things being equal, the smaller the cylinder and the lower the piston speed the greater is the value of the jacket. Experiments show no advantage in increasing the jacket pressure more than a few pounds above that of the initial steam in the cylinder, and it is usual to reduce the pressure in the jackets of the second and succeeding cylinders of multi-expansion engines. (Ripper, "Steam Engine," p. 170.)

To be effective, jackets should be well drained, kept full of live steam, and the water of condensation returned directly to the boiler.

Pumping engines and other slow-speed engines running at practically constant load are generally jacketed, but in street-railway work and in the majority of manufacturing plants carrying fluctuating load, jackets are not considered advantageous.

Whatever may be the actual economy due to jacketing, there is no question but that the jacket greatly influences the action of the steam in the cylinders, and whether beneficially or not depends upon the design and construction of the engine. Unless otherwise specified, manufacturers usually build their engines without jackets.

200. Increasing Rotative Speed. — High rotative speed does not necessarily mean high piston speed. An 8 × 10 engine running at 300 r.p.m. has a piston speed of only 500 feet per minute, whereas a 36 × 72 Corliss running at 60 r.p.m. has a piston speed of 720 feet per minute. The classification "high speed" and "low speed" refers to rotative speed only, the former above and the latter below, say 150 r.p.m.

On account of the reduction of thermodynamic wastes, a high-speed engine should give theoretically a higher efficiency than the same engine at a lower speed, all other conditions being the same. The effect of speed upon economy is decidedly marked in engines and pumps taking steam full stroke. For example, tests of a 12 × $7\frac{1}{4}$ × 12 simplex direct-acting steam pump at Armour Institute of Technology showed a steam consumption of 300 pounds per i.h.p. hour at 10 strokes per minute, and only 99 pounds at 100 strokes per minute. (See Figs. 383 and 384.)

Tests of engines using steam expansively, however, do not furnish conclusive evidence on this point, some showing a decided gain (Peabody, "Thermodynamics," p. 425), others little or no gain (Barrus, "Engine Tests," p. 260). For example, a small Willans engine showed an increase in economy of 20 per cent in increasing the rotative speed from 111 to 408 r.p.m. (Peabody, "Thermodynamics," p. 402), whereas the compound locomotive at the Louisiana Purchase Exposition showed a loss in economy for the higher speeds (Publication by the Pennsylvania Railroad Company). On the other hand, a comparison of the performances of high- and low-speed Corliss engines shows little difference in economy, and a general comparison between high- and low-speed engines furnishes little information, since nearly all high-speed engines are of a different class from the low-speed ones. High-speed engines are comparatively small in size, require larger clearance volume, and are usually fitted with a single valve. Rotative speed is limited by design, material, workmanship, and cost of subsequent maintenance. Speeds of 400 r.p.m. and more are not unusual with single-acting engines, whereas 300 r.p.m. is about the limit for double-acting machines with strokes over 12 inches in length. A comparison of tests of high-speed and low-speed engines in this country, irrespective of design and construction, shows the former to be less economical than the latter in

most cases. In Europe high-speed engines are developed to a high degree of efficiency, and their performances are comparable with the best grade of low-speed engines.

High-speed engines as a class have the advantage of being more compact for a given power, are simple in construction and relatively low in first cost; on the other hand, they are subject to comparatively rapid depreciation, excessive vibration, and are less economical in steam consumption.

201. High-speed Single-valve Simple Engines. — This style of engine is made in sizes varying from 10 to 500 horse power. The cylinder dimensions vary from 4 × 5 to 24 × 24 and the rotative speed from 300 to 175 r.p.m.

When ground is limited or costly and exhaust steam is necessary for heating or manufacturing purposes, the high-speed non-condensing engine is most suitable for horse powers of 200 or less, being compact, simple in construction and operation, and low in first cost. For sizes larger than this the compound engine would probably prove a better investment, except in cases where fuel is very cheap or large quantities of exhaust steam are to be used for manufacturing purposes.

Small high-speed engines are seldom operated condensing, since the gain due to reduction of back pressure is more than offset by the extra cost of the condenser and appurtenances.

Engines are ordinarily rated at about 75 per cent of their maximum output. For example, a 12 × 12 non-condensing engine running at 300 r.p.m., with initial steam pressure of 80 pounds gauge, is normally rated at 70 horse power, though it is capable of developing 90 horse power at the same speed.

The steam consumption of high-speed single-valve non-condensing engines at full load ranges from 27 to 50 pounds per indicated horse-power hour, depending upon the size of the unit and the conditions of operation. An average for good practice is not far from 30 pounds. With superheated steam a steam consumption as low as 18 pounds per horse-power hour has been recorded.

Table 59 gives the steam consumption of a number of single-valve high-speed engines running condensing and non-condensing, and Fig. 202 shows some of the results for different loads. The steam consumption is fairly constant from 50 per cent of the rated load to 25 per cent overload, but for earlier loads the economy drops off rapidly. The desirability of operating the engine near its rated load is at once apparent. The curves show a marked economy in favor of the larger cylinders, but the engines are not of the same make, and the conditions of operation are somewhat different.

The most economical cut-off for a simple engine is about one-third to one-fourth stroke when running non-condensing, and about one-sixth when running condensing.

The performances given in Table 59 are exceptional. It is not advisable to count on a better steam consumption for this type of engine than 30 to 35 pounds of steam per i.h.p. hour.

Fig. 206 shows the effects of condensing on a typical single-valve high-speed engine. The gain in fuel economy may be only an apparent one, since the steam consumption of the condensing apparatus should be rightfully charged to the engine.

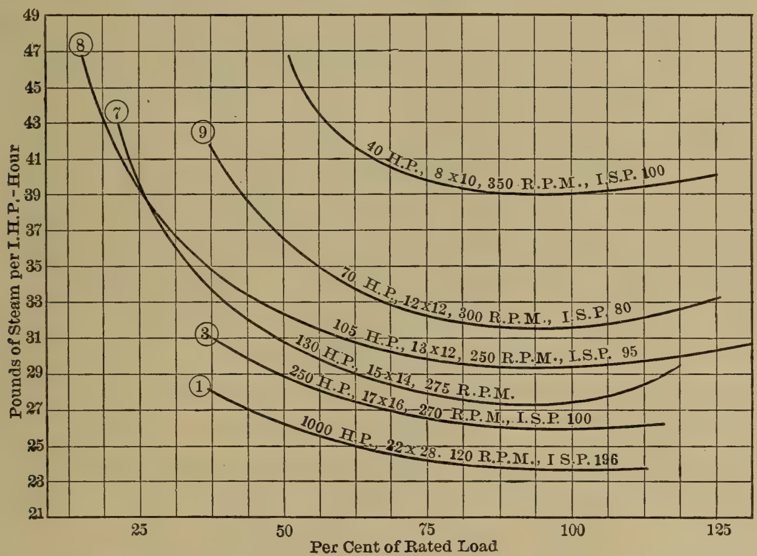


FIG. 202. Typical Economy Curves of High-speed, Single-valve, Non-condensing Engines. Saturated Steam.

Fig. 207 shows the relation between total steam supplied, unit water rate and quality of exhaust at various loads for a 22 × 18 high-speed, single-valve, non-condensing engine direct connected to a 200-kilowatt direct-current generator as installed in the power plant of the Armour Institute of Technology.

These curves afford a means of determining the heat actually required to furnish power when the exhaust is used for heating purposes. During the summer months the total heat supplied measured above the feed-water temperature is chargeable to power. During the winter months the heat required to furnish the electrical energy is the difference between the total heat supplied and that exhausted to the heating system. (See paragraph 194.) This latter is readily obtained from the quality

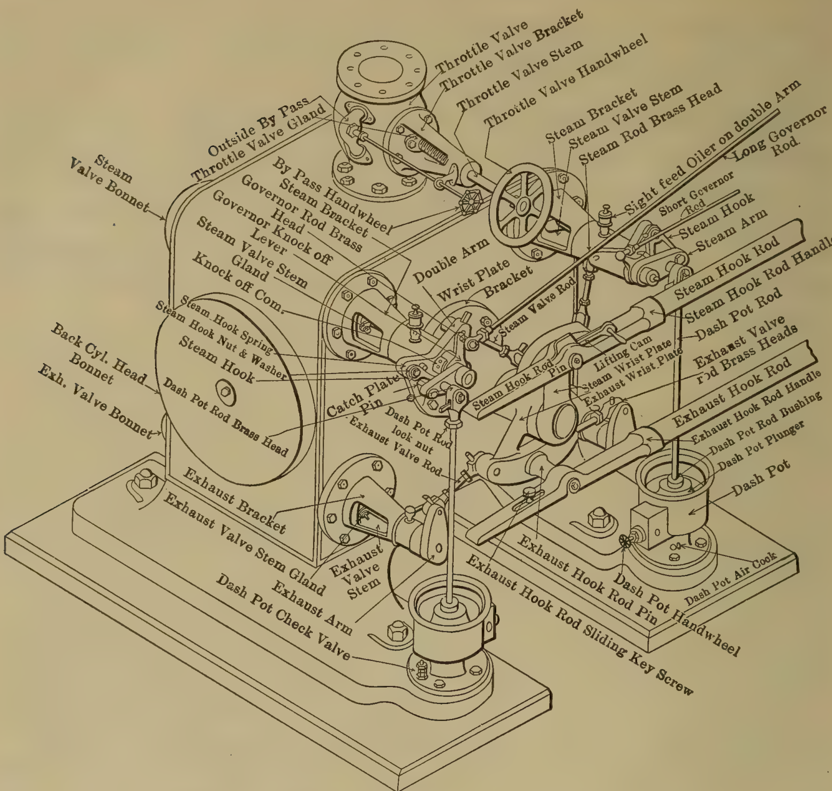


FIG. 203. Assembly of Valve Gear; Typical Corliss Engine.

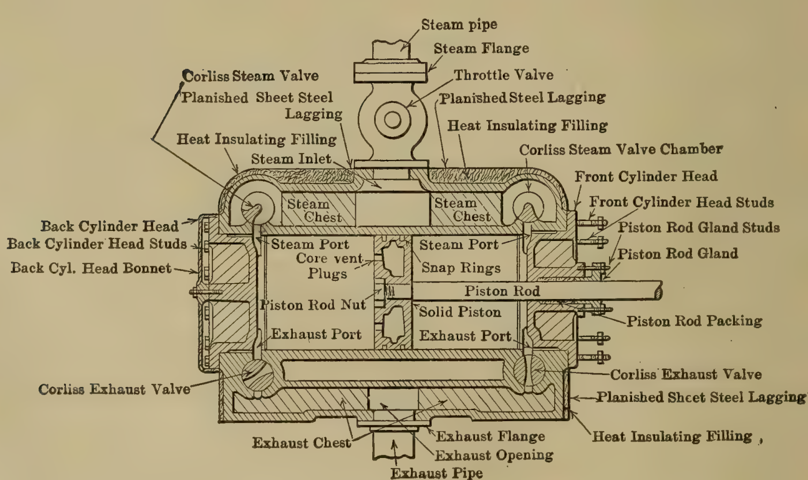


FIG. 204. Section Through Cylinder; Typical Corliss Engine.

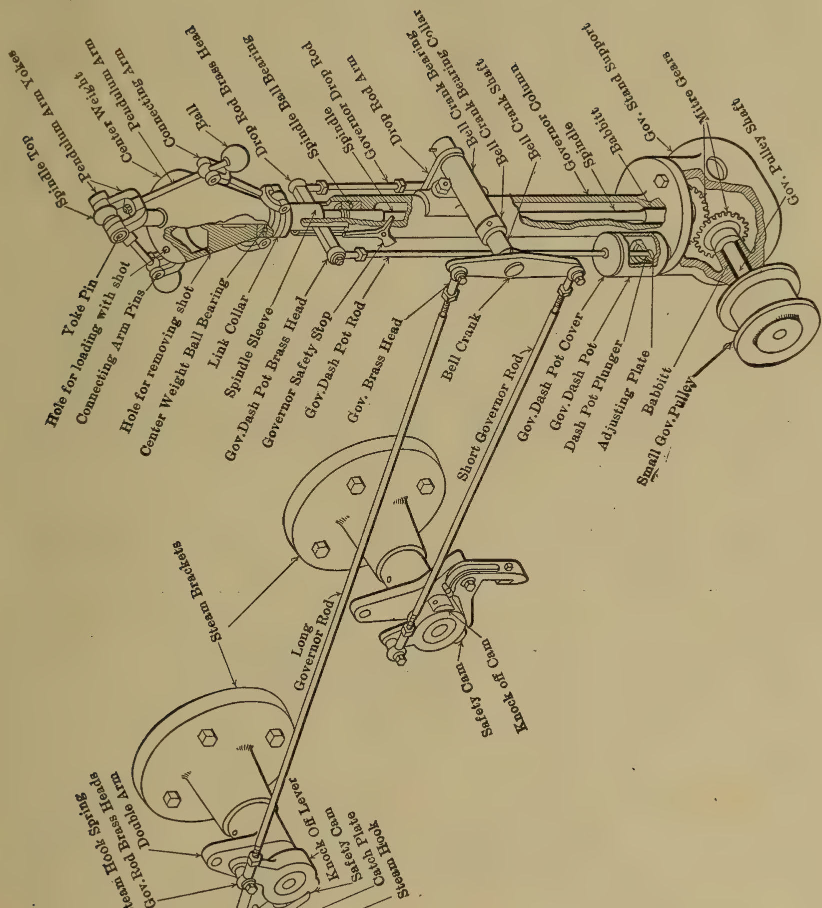


FIG. 205. Assembly of Governor and Link Mechanism; Typical Corliss Engine.

curve. By means of an indicating or recording flow meter placed in the exhaust line leading to the heating system the heat chargeable to power may be readily obtained during the months when only part of

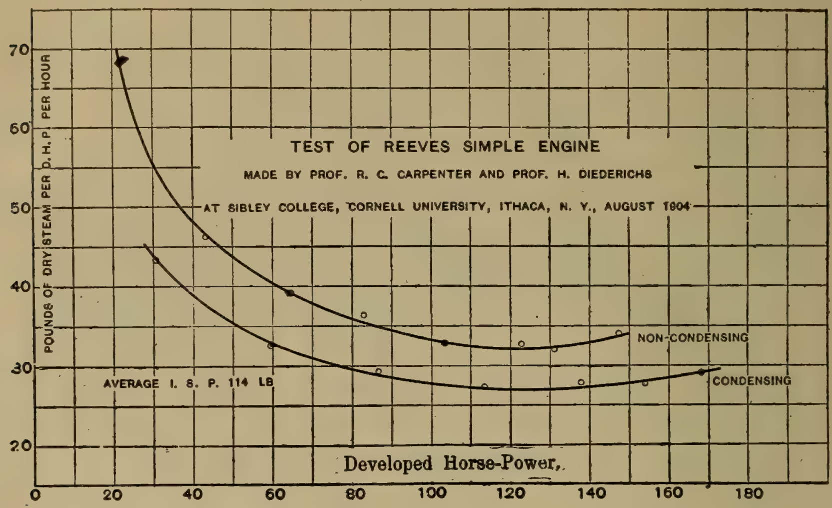


Fig. 206.

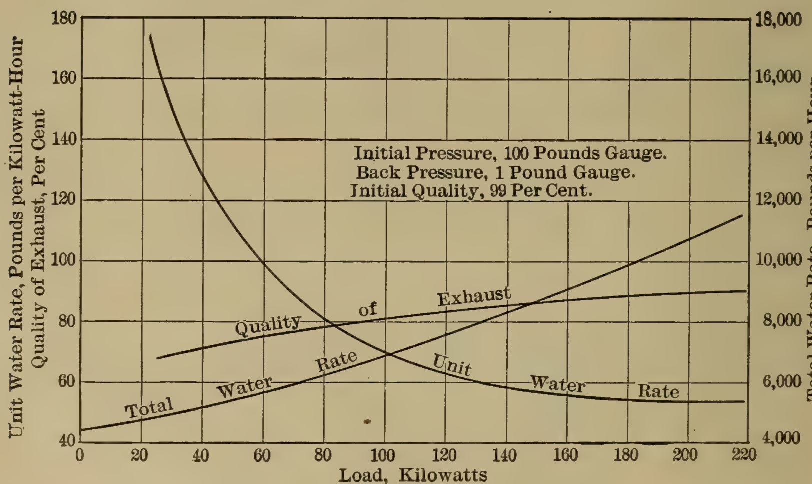


Fig. 207. — Performance of 200-kilowatt Direct-connected Engine-generator Set. Single, Non-condensing Engine.

the exhaust is used for heating purposes. Although the quality curves refer to a special case they may be used as a means of roughly approximating the heat exhausted by any high-speed non-condensing engine.

In general, when the requirements for exhaust steam are in excess of the steam consumption of a simple non-condensing engine a high-grade economical engine is without purpose.

TABLE 58.

ECONOMY OF AUTOMATIC SINGLE-CYLINDER NON-CONDENSING ENGINES.

Indicated Horse Power.	Pounds Steam per Indicated Horse Power Hour at			
	Full Load.	Three-quarter Load	One-half Load.	One-quarter Load.
80	29.42	29.93	31.66	37.08
100	28.96	29.40	31.04	36.00
125	28.47	28.84	30.42	35.10
150	28.12	28.46	29.95	34.38
200	27.51	27.81	29.25	33.26
300	26.64	26.75	27.97	31,68

Above engine economies are based on dry saturated steam at 125 lbs. Engine horse powers are figured with steam cutting off at 25% of stroke. The economies given are for medium speed piston valve engines of the highest type only.

202. High-speed Multi-valve Simple Engines. — The steam distribution in a single-valve engine may give good economy for a very small range in load but be far from satisfactory for a wide range. This must necessarily be so since admission, cut-off, release, and compression are all functions of one valve, and any change in one results in a change of the others. To obviate the limitations of the single valve, many builders design engines with two or more valves. With a two-valve engine cut-off is independent of the other events, and with four valves all events are independently adjustable. In addition to the flexibility of the valve gear, the chief feature of the four-valve engines lies in the reduction of clearance volume which is made possible by placing the valves directly over the ports. The valves may be of the common slide-valve or rotary type. As a class, four-valve engines are more economical than those having a less number of valves. The advantages and disadvantages of the four-valve over the single-valve engines may be tabulated as below.

ADVANTAGES.

1. Better steam distribution.
2. Better regulation.
3. Reduced clearance volume.
4. Less valve leakage.
5. Better economy.

DISADVANTAGES.

1. Increased number of parts.
2. Increased first cost.
3. Requires greater attention.

The steam consumption of a high-speed four-valve non-condensing engine varies from 22 to 35 pounds of saturated steam per horse-power

TABLE 59. — EXAMPLES OF STEAM-ENGINE ECONOMY.

SIMPLE ENGINES, SATURATED STEAM.

Index.	Kind of Engine.	References.	Cylinder Dimensions.	I.H.P.	Initial Pressure, Lbs. Gauge.	Back Pressure, Lbs. per Sq. In., Abs.	M.E.P.	R.P.M.	Lbs. of Dry Steam per I.H.P. Hour.	B.T.U. per I.H.P. per Minute.	B.T.U. per I.H.P. per Minute, Perfect Engine.	Efficiency Ratio, per Cent.	Mechanical Efficiency, per Cent.
		SINGLE-VALVE, NON-CONDENSING.											
1	Willans	Peabody, Thermodynamics	14 × 6	33.6	122.0	Atmo.	....	400	26.0	436	263	60.4	....
2	Willans	Peabody, Thermodynamics	14 × 6	16.5	36.3	..do..	....	393	42.8	704	570	81.0	....
3	Locomotive, Purdue	Trans. A.S.M.E., Vol. 14, p. 826	17 × 14	399.0	110.0	..do..	54.0	136	24.97	420	273	65.0	....
4	Westinghouse Standard	Shop test	20 × 16	257.0	100.0	..do..	36.0	275	26.19	439	286	65.2	....
5	Buffalo forge	Elec. World, Sept. 1, 1904, p. 407	12 × 12	121.0	124.0	..do..	58.5	302	27.5	462	265	57.5	....
6	Reeves	Elec. World, Oct. 1, 1904, p. 587	15 × 14	120.0	114.0	..do..	35.0	275	28.0	470	270	57.5	92.5
7	Ames	Eng. Record, July 6, 1901, p. 7	13 × 12	105.0	95.0	..do..	52.0	250	29.1	493	293	59.4	....
8	Ames	Meyer, Steam Power Plants, p. 56	17 × 16	248.0	100.0	..do..	50.0	270	26.0	435	286	65.7	....
9	Locomotive, No. 1499, Penn. System.	Locomotive Tests, 1904, at Louisiana Exposition.	22 × 28	975.0	196.0	..do..	75.6	120	23.4	398	222	55.9	91.0
10	Buffalo forge	Elec. World, Sept., 1904, p. 407	12 × 12	74.0	79.3	..do..	35.6	304	30.6	510	324	63.5	9.57
		SINGLE-VALVE, CONDENSING.											
11	Willans	Peabody, Thermodynamics	14 × 6	33.2	70.0	1.0	....	383.0	22.2	410	169	41.3	....
12	Reeves	Elec. World, Oct. 1, 1904, p. 587	15 × 14	140.0	114.0	3.2	41.0	275.0	26.0	466	186	39.7	95.8
13	Buffalo forge	Elec. World, Sept. 10, 1904, p. 407	12 × 12	86.0	80.0	3.0	40.5	310.0	27.5	490	197	40.0	95.0
14	Piston valve	Barrus, Engine Tests, p. 95	18½ × 30	204.6	69.3	2.6	38.1	29.3	27.15	435	199	41.0	....
		FOUR-VALVE, NON-CONDENSING.											
15	Corliss, jacketed	Peabody, Thermodynamics	21.6 × 43.3	237.0	103.5	Atmo.	42.1	62.7	21.5	358	280	78.0	....
16	Fleming	Prof. Carpenter, June 28, 1905, at Cornell University.	19 × 19.0	217.9	120.5	..do..	39.0	205.9	22.46	378	265	70.0	....
17	Fleming	Prof. Spangler, June 6, 1905, at University of Pennsylvania.	16 × 16.0	132.0	125.4	..do..	39.1	210.0	22.24	374	260	69.5	95.0
18	Corliss	Peabody, Thermodynamics	18 × 48.0	120.0	96.0	..do..	25.7	76.0	23.9	400	292	73.5	....
19	Corliss	Barrus, Engine Tests, p. 47	28½ × 59½	506.0	101.0	..do..	41.0	64.8	25.8	432	286	66.2	....
20	Corliss	Barrus, Engine Tests, p. 126	16 × 42.0	342.0	91.6	..do..	48.4	84.9	25.9	434	312	72.0	....
		FOUR-VALVE, CONDENSING.											
21	Corliss, jacketed	Peabody, Thermodynamics	21.6 × 43.3	155.0	103.8	1.2	32.0	60.0	16.5	302	161	53.3	....
22	Poppet valves, jacketed	Zeit. d. V. D. Ing., Aug., 1905, p. 1310.	22.6 × 45.0	262.0	79.0	1.36	30.2	47.6	15.0	275	175	63.8	....
23	Gridiron valves	Barrus, Engine Tests, p. 101	34½ × 60.0	613.0	82.3	1.0	37.2	60.0	18.5	342	164	48.0	....
24	Corliss	Barrus, Engine Tests, p. 118	32 × 60.0	554.0	67.5	2.9	38.2	59.1	19.45	342	206	60.2	....
25	Slide valves	Barrus, Engine Tests, p. 88	18 × 30.0	213.0	67.0	2.2	33.7	165.0	22.0	397	195	49.1	....
26	Corliss	Peabody, Thermodynamics	18 × 48.0	145.0	96.0	1.5	30.9	76.0	19.4	356	171	48.0	....

hour, with an average not far from 27 pounds. With superheated steam the steam consumption may run as low as 15 pounds per horse-power hour.

Fig. 208 gives a comparison between a single-valve and a four-valve high-speed engine, and though the engines differ slightly in size, the conditions of operation were comparable and the marked gain in economy of the latter over the former is apparent. Both performances are exceptional, and a 10 to 15 per cent greater steam consumption may be expected in average good practice.

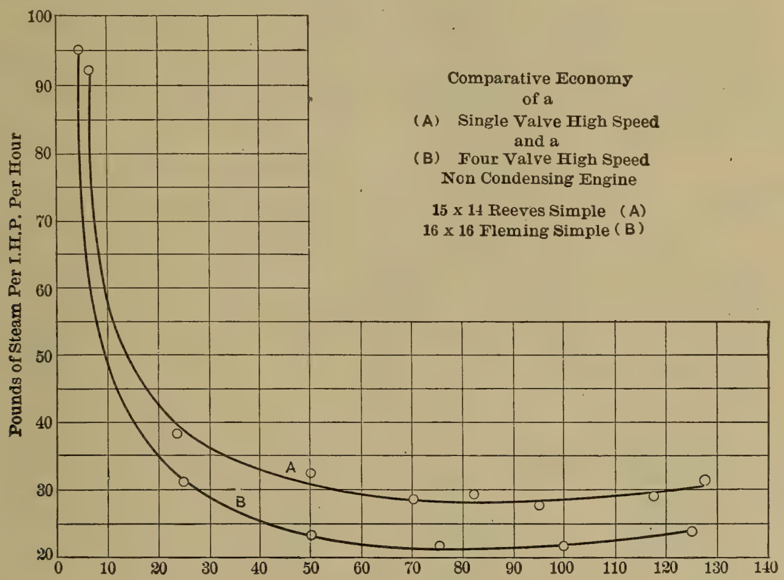


Fig. 208.

As a general rule single-valve simple engines do not exceed 500 horse power in size for stationary work, whereas 1000 horse power is not an uncommon size for the multi-valve type.

203. Medium and Low-speed Multi-valve Simple Engines. — A comparison of tests of high- and low-speed single-valve engines irrespective of design and construction shows the former as a class to be less economical than the latter. With four-valve engines there is no such disparity, and the high-speed type has shown just as good economy as the slow-speed class. For example, Engine No. 17, Table 59, with Corliss valves and a speed of 210 r.p.m., gives practically the same economy as Corliss engine No. 15 operating at 62 r.p.m. By far the greater number of simple multi-valve slow-speed simple engines are of the Corliss

type. They range in size from 50 to 3000 horse power, with cylinders varying from 12 × 30 to 48 × 72. The smaller sizes with trip-valve gear run at 90 to 100 r.p.m., and the larger at 50 to 75 r.p.m. Without the trip gear, speeds of 150 r.p.m. are not uncommon, but at this speed they are usually classified as high-speed engines.

Table 60 gives the steam consumption, condensing and non-condensing, of a number of four-valve slow-speed simple engines. For performances of uniflow engines, see paragraph 210.

TABLE 60.

ECONOMY OF CORLISS OR MEDIUM-SPEED FOUR-VALVE SIMPLE ENGINES.

Following engine economies are based on dry saturated steam at 130 pounds and exhausting against atmospheric pressure. Engine horse powers are figured with steam cutting off in the cylinder at 25 per cent of stroke.

Indicated Horse Power.	Pounds Steam per Indicated-horse-power Hour at			
	Full Load.	Three-quarter Load	One-half Load.	One-quarter Load.
200	23.45	23.05	24.75	35.00
350	23.03	22.54	24.07	33.79
500	22.61	22.06	23.45	32.73
650	22.24	21.67	22.92	31.71
800	22.00	21.40	22.57	30.91
900	21.90	21.31	22.44	30.48

204. Compound Engines.— Compound engines may be divided into three classes, tandem, cross compound, and duplex. In the tandem the two cylinders are end to end, in the cross compound side by side, and in the duplex one above the other. The tandem and duplex compounds have the advantage of (1) compactness for a given power, (2) less complication and fewer parts, and (3) low first cost. The crank effort is more variable than in the cross compound. In very large engines the low-pressure stage is generally divided between two cylinders of equivalent size to avoid an excessively large single cylinder and to distribute the crank effort. High-speed non-condensing compounds are ordinarily of the tandem type and are finding much favor in isolated station work, as in the power plants of tall office buildings where ground space is limited, though the duplex compound is sometimes used. The vertical or horizontal cross compound is generally installed in street-railway plants.

Cylinder ratios for high-speed single-valve compound engines vary from about 1 to $2\frac{1}{2}$ with 100 pounds pressure to about 1 to 3 with a pressure of 150 pounds, and for slow-speed condensing engines from 1 to 3 with 125 pounds pressure to about 1 to 4 with a pressure of 175 pounds.

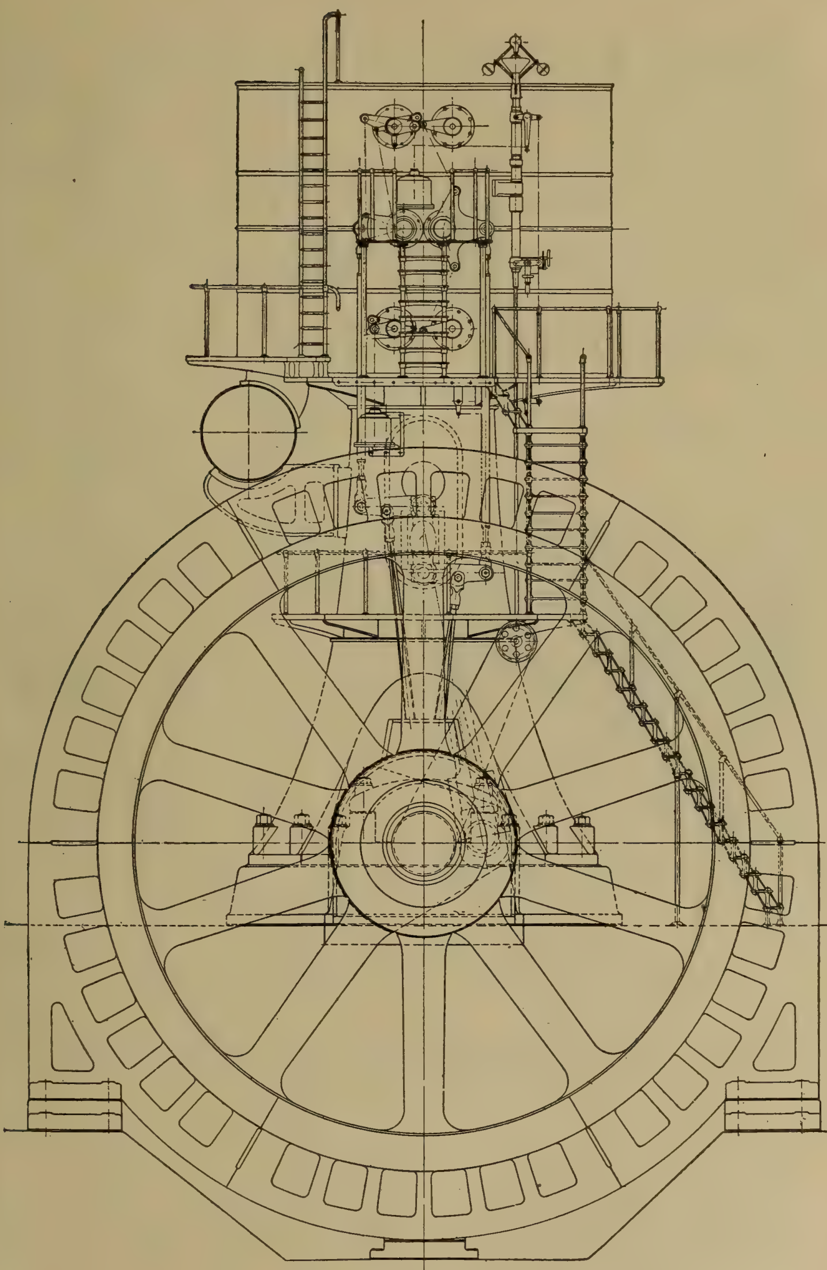


FIG. 209. 3500 K.W. Vertical Cross-Compound Corliss Engine as Installed at the Power House of the Twin City Rapid Transit Co., Minneapolis, Minn.

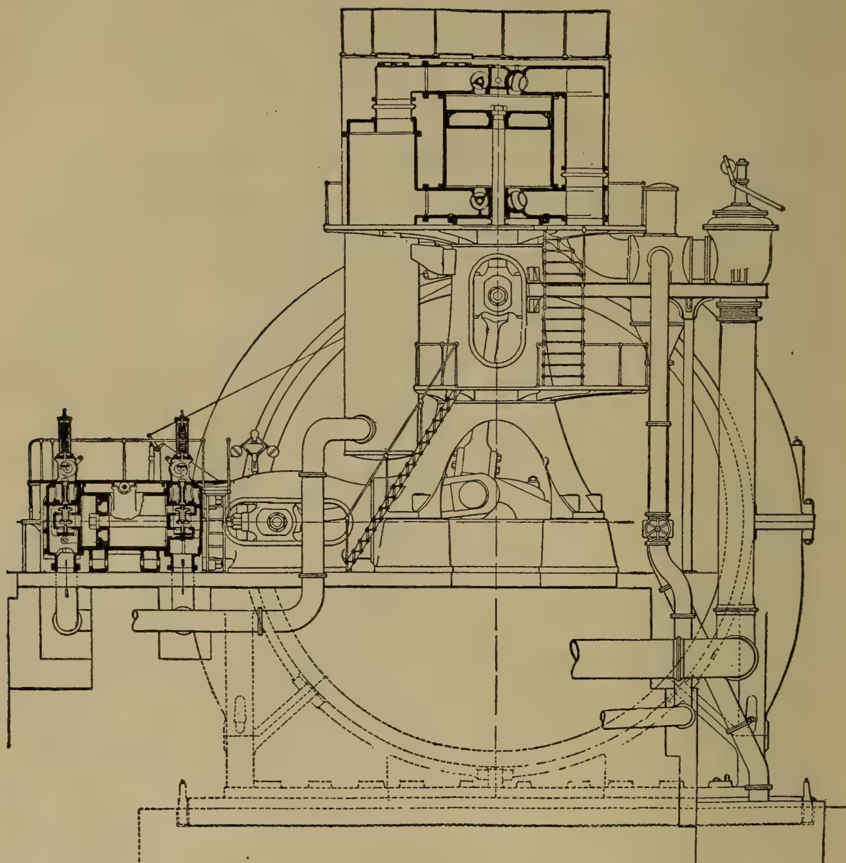


FIG. 210. 7500 K.W. Vertical-horizontal Double-compound Engine as Installed at the 59th Street Station of the Interborough Rapid Transit Co. (Manhattan Type.)

G. I. Rockwood recommends a ratio as high as 7 to 1, and a number of engines designed along this line have shown exceptional economy. A cross-compound Corliss engine at the Atlantic Mills, Providence, R.I., with cylinders 16 and 40 × 48 (ratio 6.128 to 1) gave the low steam consumption of 11.2 pounds of steam per i.h.p. hour, corresponding to a heat consumption of 222 B.t.u. per i.h.p. per minute. The 5500-horse-power engines of the New York Edison Company have a cylinder ratio of 6 to 1. The great majority of compound engines, however, have cylinder ratios of 4 to 1 or less. The 8000-horse-power engines of the Interborough Rapid Transit system have a ratio of 4 to 1, and the 4000-horse-power units of the Metropolitan Elevated Company, New York, a ratio of 3.5 to 1.

The respective advantages and disadvantages of compounding may be tabulated as follows:

Advantages.

1. Permits high range of expansion.
2. Decreased cylinder condensation.
3. Decreased clearance and leakage losses.
4. Equalized crank effort.
5. Increased economy in steam consumption.

Disadvantages.

1. Increased first cost due to multiplication of parts.
2. Increased bulk.
3. Increased complexity.
4. Increased wear and tear.
5. Increased radiation loss.

The ratio of expansion for a multi-expansion engine is usually taken to be the product of the ratio of the volume of large to small cylinder divided by the fraction of the stroke at cut-off in the high-pressure cylinder. For example, a compound engine with cylinders 24, 48 × 48 cutting off at $\frac{1}{3}$ in the high-pressure cylinder has a nominal ratio of expansion of $4 \div \frac{1}{3} = 12$. The number of expansions at rated load in compound condensing engines varies widely, ranging from 10 to 33, with an average not far from 16.

The steam consumption shown by tests of a number of compound engines using saturated steam, condensing and non-condensing, is given in Table 66. For tests with superheated steam see Table 69.

Fig. 211 shows the relative economy under comparable conditions of a high-speed simple and a high-speed compound engine, both running non-condensing and using saturated steam. The advantage of the compound at full load and overload is very marked, though its economy drops off rapidly at light loads and may be less than that of the simple engine.

Fig. 212 shows the relative economy of two compound Corliss engines running condensing and non-condensing, both using saturated steam.

It should be borne in mind that the object of compounding is to permit the advantageous use of high pressures and large ratios of ex-

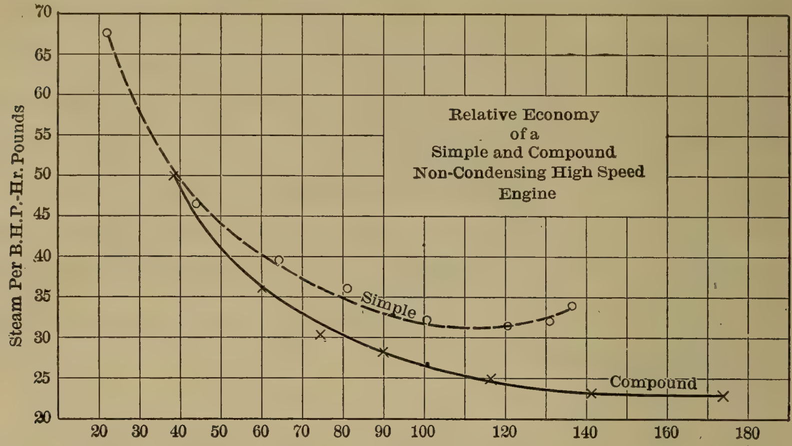


FIG. 211.

pansion. Under proper conditions compounding may increase the economy at rated load about 20 per cent for non-condensing engines and 30 per cent for condensing engines.

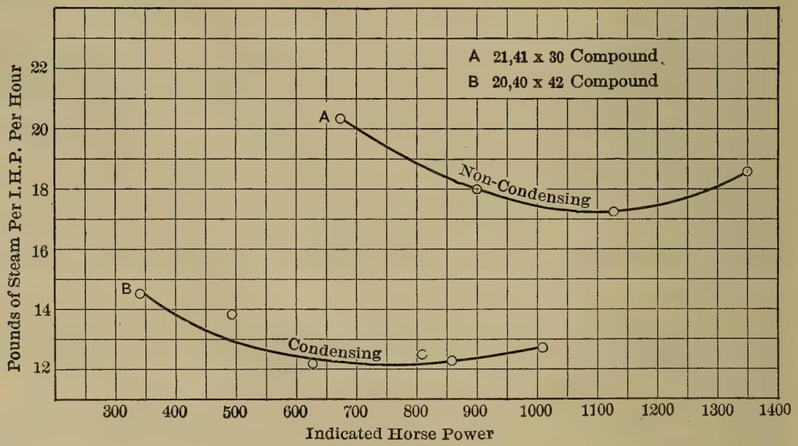


FIG. 212.

An exceptional performance of a single-valve high-speed non-condensing compound engine is that of engine No. 20, Table 66. With initial gauge pressure of 128 pounds the steam consumption is 22.3 pounds per i.h.p. hour, corresponding to a heat consumption of 376 B.t.u. per i.h.p. per minute.

One of the best performances of a multi-valve high-speed compound non-condensing engine is that of engine No. 14, Table 66. With initial pressure of 175 pounds gauge the steam consumption at full load is 17.17 pounds per i.h.p. hour, corresponding to a heat consumption of 291 B.t.u. per i.h.p. per minute.

The 8000-horse-power vertical cross-compound Corliss engines of the Interborough Rapid Transit system (No. 6, Table 66) probably hold

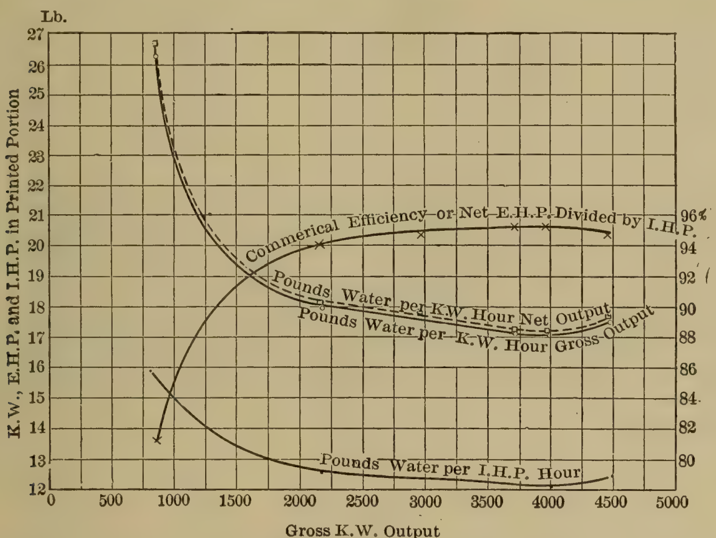


FIG. 213. Economy Test of the 5500-Horse-power Three-cylinder Compound Engine and Generator at the Waterside Station of the New York Edison Co.

the record for economy for compound engines without jackets and reheaters, using saturated steam. With initial pressure of 175 pounds gauge and absolute back pressure of 2.2 pounds, the steam consumption is 11.96 pounds per i.h.p. hour, corresponding to a heat consumption of 220 B.t.u. per i.h.p. per minute. In estimating average practice it would be safe to add 10 per cent or 20 per cent to the steam consumptions given in Table 66.

Fig. 213 illustrates the performance of the 5500-horse-power three-cylinder compound engine at the Waterside Station of the New York Edison Company. The best economy is 11.93 pounds of steam per i.h.p. hour, corresponding to a heat consumption of 221 B.t.u. per i.h.p. per minute.

TABLE 61.

ECONOMY OF AUTOMATIC TANDEM COMPOUND ENGINES RUNNING NON-CONDENSING.

Indicated Horse Power.	Pounds Steam per Indicated Horse Power at —			
	Full Load.	Three-quarter Load.	One-half Load.	One-quarter Load.
100	24.04	25.06	29.54	43.84
150	22.94	23.82	28.00	41.40
200	22.36	23.19	27.19	40.46
250	21.98	22.75	26.65	39.13
350	21.48	22.18	25.96	38.01
450	21.27	21.92	25.61	37.45

Above engine economies are based on dry saturated steam at 140 pounds. Cylinder ratios are 4 to 1, and engine horse powers are figured with steam cutting off in the high-pressure cylinder at 25 per cent of stroke. The economies given are for medium-speed piston-valve engines of the highest type only.

Above cylinder ratios are for engines normally operating condensing. These economies are given, however, so that engineers may know the steam consumption of this class of engine when it becomes necessary to operate same non-condensing, through lack of condensing water, or when it is desired to use the exhaust steam for heating purposes.

TABLE 62.

ECONOMY OF AUTOMATIC TANDEM COMPOUND CONDENSING ENGINES.

Following engine economies are based on dry saturated steam at 140 pounds, and vacuum of 26 inches. Cylinder ratios are 4 to 1, and engine horse powers are figured with steam cutting off in the high-pressure cylinder at 25 per cent of stroke. The economies given are for medium-speed piston valve engines of the highest type only.

Indicated Horse Power.	Pounds Steam per Indicated Horse-power Hour at —			
	Full Load.	Three-quarter Load.	One-half Load.	One-quarter Load.
150	20.25	21.51	26.00	37.83
300	19.10	20.12	24.10	33.90
400	18.55	19.45	23.11	32.07
500	18.15	18.93	22.44	30.90
600	17.92	18.67	22.08	30.20
700	17.83	18.55	21.91	29.95

TABLE 63.

ECONOMY OF CORLISS OR MEDIUM-SPEED FOUR-VALVE TANDEM COMPOUND ENGINES RUNNING NON-CONDENSING.

Indicated Horse Power.	Pounds Steam per Indicated Horse-power Hour at —			
	Full Load.	Three-quarter Load.	One-half Load.	One-quarter Load.
300	19.71	21.74	26.82	39.54
450	19.20	21.10	25.90	37.80
600	18.91	20.66	25.20	35.46
750	18.74	20.41	24.73	35.31
850	18.68	20.32	24.55	34.81
950	18.66	20.30	24.48	34.47

Above engine economies are based on dry saturated steam at 150 pounds pressure and exhausting at atmospheric pressure. Cylinder ratios are 4 to 1, and engine horse powers are figured with steam cutting off in the high-pressure cylinder at 25 per cent of stroke.

Above cylinder ratios are for engines normally operating condensing. These economies are given, however, so that engineers may know the steam consumption of this class of engine when it becomes necessary to operate same non-condensing, through lack of condensing water, or when it is desired to use the exhaust steam for heating purposes.

TABLE 64.

ECONOMY OF CORLISS OR MEDIUM-SPEED FOUR-VALVE TANDEM COMPOUND CONDENSING ENGINES.

Indicated Horse Power.	Pounds Steam per Indicated Horse-power Hour at —			
	Full Load.	Three-quarter Load.	One-half Load.	One-quarter Load.
300	15.42	15.30	17.26	24.00
500	14.74	14.60	16.45	22.47
700	14.29	14.10	15.78	21.30
900	13.97	13.76	15.34	20.48
1100	13.73	13.51	15.02	19.94
1300	13.56	13.33	14.83	19.66
1500	13.49	13.23	14.71	19.52

Above engine economies are based on dry saturated steam at 160 pounds and 26 inches vacuum. Cylinder ratios are 4 to 1 and engine horse powers are figured with steam cutting off in the high-pressure cylinder at 25 per cent of stroke.

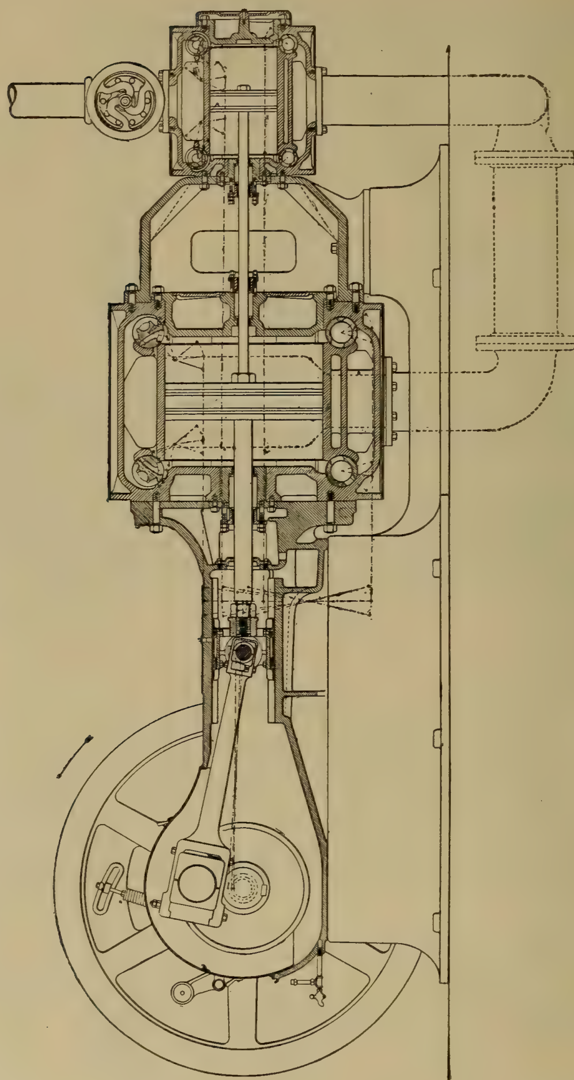


FIG. 214. Fleming-Harrisburg Four-valve High Speed Tandem Compound Engine.

205. Triple and Quadruple Expansion Engines. — Triple and quadruple expansion engines are still in use where the load is practically constant, as in marine and pumping-station practice, but have been abandoned in street-railway work and in plants where the load fluctuates widely

TABLE 65.

ECONOMY OF MODERN VERTICAL TRIPLE-EXPANSION PUMPING ENGINES.

(Official Trials.)

Date of Test.	Type.	Location.	Rated Capacity, Millions of U.S. Gallons.	Initial Gauge Pressure.	Duty.		Dry Steam per I.h.p. Hour.
					Per Thousand Lbs. of Dry Steam.	Per One Million B.t.u.	
5- 2-09	Holly	Louisville, Ky.......	24	155.1	*195.0	164.5	*9.64
3-10-10	Holly	Frankfort, Pa.......	20	180.2	184.4	......	......
4-29-10	Holly	Albany, N. Y.......	12	153.0	182.1	......	......
10-14-09	Holly	Brockton, Mass.....	6	150.0	170.0	......	......
12- 5-07	Holly	Cleveland, Ohio.....	2.5	149.6	164.6	148.8	11.51
5- 2-00	Allis	Boston, Mass.......	30	185.5	178.5	163.9	10.33
2- 4-06	Allis	St. Louis, Mo.......	20	140.6	181.3	158.8	10.66
2-26-00	Allis	St. Louis, Mo.......	15	126.2	179.4	158.1	10.67
1-15-10	Allis	Milwaukee, Wis.....	12	124.6	175.4	151.0	10.82

* 109 degrees F. superheat at throttle.

Date of Test.	Type.	R.P.M.	Water Actually Pumped, Millions of U.S. Gallons 24 Hrs.	Net Head Pumped Against, Lbs. per Sq. In.	Indicated Horse Power.	Developed Horse Power.	Thermal Efficiency Per Cent. I.h.p.
5- 2-09	Holly	24.0	24.111	90.0	925.7	879.4	22.54
3-10-10	Holly	20.1	21.219	95.7	.....	817.0	.....
4-29-10	Holly	22.3	12.193	139.5	.....	726.0	.....
10-14-09	Holly	40.1	6.316	130.6	.....	334.0	.....
12- 5-07	Holly	62.3	2.142	180.7	158.7	151.9	19.13
5- 2-00	Allis	17.7	30.314	61.0	801.5	747.8	21.63
2- 4-06	Allis	16.5	20.070	104.0	859.2	839.6	20.92
2-26-00	Allis	16.4	15.121	127.0	801.6	726.3	21.00
1-15-10	Allis	20.4	12.430	121.0	673.0	618.0	20.25

in favor of the two- or three-cylinder compound. The best economy on a *heat-unit* basis ever recorded for an engine using saturated steam was that of the Nordburg quadruple pumping engine at Wildwood, Pa., which gave a consumption of 12.26 pounds per i.h.p. hour and a heat consumption of 186 B.t.u. per i.h.p. per minute reckoned above the feed-water temperature.* The Allis triple-expansion pumping engine

* Replaced in 1905 by a Riedler pumping engine on account of high maintenance cost.

TABLE 66.—EXAMPLES OF STEAM-ENGINE ECONOMY.

MULTIPLE EXPANSION ENGINES, SATURATED STEAM.

Index.	Kind of Engine.	References.	Cylinder Dimensions.	Cylinder Ratio.	Horse Power.	Initial Press., Lbs. Gauge.	Back Press., Lbs. Abs.	R.P.M.	M.E.P. Referred to L.P. Cyl.	Temp. Feed Water, Deg. F.	Lbs. of Steam per I.H.P. Hr.	B.T.U. per I.H.P. per Min.	Thermal Efficiency, per Cent.	B.T.U. per I.H.P. Min., Perfect Eng.	Efficiency Ratio, per Cent.	Mech. Efficiency, per Cent.	Lbs. of Coal per D.H.P. Hr.
			Quadruple Expansion.														
1	Nordburg Pumping Engine, Wildwood, Pa.	Eng. News, May 4, 1899, p. 280.	19½, 29, 49½, 57½ x 42		712	200	0.9	36.5	35.5	310.8	12.26	186	22.8	138	74.2	93.0	1.12
			Triple Expansion.														
2	Allis Pumping Engine, Chestnut Hill, Boston.	Eng. News, Aug. 23, 1900, p. 125.	30, 56, 87 x 66	1:3½:8.4	801	185	0.85	17.2	23.4	155	10.33	196	21.63	138	70.5	*93.3	1.09
3	Allis Pumping Engine, Bissell's Point, St. Louis.	Power, May 1906, p. 299.	34, 62, 94 x 72	1:3.3:7.6	865	140	1.2	16.5	20.8		10.59	201.4	21.06	151	75.0	*97.4	
4	Holly Pumping Engine, Spot Pond, Boston.	Eng. News, Nov. 14, 1901, p. 371.	22, 41, 62 x 60	1:3½:8	464	150	1.05	24.8	20.5	157.9	11.01	203.4	20.85	142	70.0	*96.5	1.11
5	Sulzer Mill Engine, Augsburg.	Zeit. d. V.D.I., May 16, 1896, p. 534.	29.9, 44.5, 2(51.6) x 78.7	1:2.2:5.9	1823	134	1.8	56.2	19.5	122	11.33	208	20.40	158	76.0		1.19
			Compound, Condensing.														
6	Allis-Chalmers Engines, New York Subway.	Power, Feb., 1906, p. 115.	2(42), 2(86) x 60	1:4.2	7365	175	2.2	75	27.9	130	11.96	220	19.2	159	72.4	†93.0	
7	Cross-Compound Corliss, Atlantic Mills, Providence.	Am. Elecn., June, 1903, p. 260.	16, 40 x 48	1:6¼	500	170	0.8	80	20.5		11.20	222	19.0	141	63.5		1.20
8	Leavitt Pumping Engine, Louisville, Ky.	Trans. A.S.M.E., Vol. 16, p. 169.	27, 54 x 120	1:4	643	137	0.95	18.6	24.9	100	12.20	222	19.0	150	67.6	93.0	
9	Rice & Sargent Corliss, Amer. Sugar Refinery, Brooklyn.	Trans. A.S.M.E., Vol. 24, p. 1274.	20, 40 x 42	1:4	627	151	0.85	121	19.4	121	12.10	222.7	19.0	143	64.3		
10	Fleming Four-Valve.	Trans. A.S.M.E., Vol. 25, p. 212.	15, 40½ x 27	1:7.3	348	150	2.0	152	13.0	126	12.33	225.8	18.7	162	71.7		
11	Williams Vertical, New York Navy Yard.	Power, Oct., 1903, p. 583.	19, 34 x 30	1:3¼	340	100	2.0	150	16.5	126	12.60	229	18.5	175	76.5		
12	Tandem-Compound Corliss.	Barrus, Eng. Tests, p. 185.	18, 44 x 72	1:6.4	6893	145.2	1.5	60.3	20.6	116	12.70	234	18.1	157	67.0		
13	Edison Waterside Sta., N. Y.	Power, July 1904, p. 424.	43½, 75.3 x 60	1:6.02	5442	185	1.5	76.3	26.5	116	11.93	221	19.2	150	68.0	†95.2	
			Compound, Non-Condensing.														
14	Ball & Wood Co. Corliss, W. Albany Sta., N. Y. C. & H. R. R.	Test by Company Engineers.	21, 41 x 30	1:3.8	1125	175	Atmos.	120	47.0	Assumed for Comparison, 212° F.	17.17	291	14.5	229	78.8		
15	Willans.	Peabody, Thermodynamics.	10, 14 x 6	1:2	39.6	165.0	Atmos.	401	42.4		19.2	328	12.8	234	71.5		
16	Willans.	Peabody, Thermodynamics.	10, 14 x 6	1:2	33.0	113.9	15.7	402			21.4	358	12.4	297	83.0		
17	Ball Engines, Chicago Public Library.	Eng. Record, Aug. 6, 1898, p. 206.	12, 20 x 13	1:2.8	187.5	166.8	Atmos.	271	34.8		21.14	357	11.7	233	65.5	93.5	
18	Westinghouse Marine.	Power, Aug., 1903.	17, 27 x 24	1:2¼	540	148	Atmos.	210	37.1		19.3	326	13.0	244	75.0	†87.5	
19	Skinner Cross-Compound.	Power, July, 1906.	16, 27 x 18	1:2.84	375	130	Atmos.	226	31.0		21.14	355	11.9	255	72.0	91.5	
20	Buffalo Tandem-Compound.	Elec. World, May 23, 1903, p. 897.	12, 18 x 10	1:2¼	121	128	Atmos.	271	35.0		22.3	376	11.2	258	68.5	93.0	
21	Reeves Vert. Cross-Compound.	Eng. Record, July 1, 1905, p. 24.	12, 20 x 14	1:2.77	185	150	Atmos.		56.0		20.9	354	11.9	242	68.5	92.0	
22	Cross-Comp'd, 4 Slide Valves.	Barrus, Eng. Tests, p. 181.	17½, 28 x 48	1:2.58	486.7	125.3	Atmos.	99	32.9		21.59	362	11.7	260	71.9		
23	4-Cylinder Compound Locomotive, No. 2512 Penn. System.	Tests made at Louisiana Exposition, 1904.	14.2, 23.7 x 25.2	1:2.78	495	210	Atmos.	80	55.0		18.6	316	13.4	216	68.5	91.0	

* Combined efficiency of pump and engine. † Combined efficiency of engine and generator.

at Chestnut Hill holds the record for saturated-steam consumption, 10 pounds per i.h.p. hour, and its exceptional performance of one developed horse power per 1.09 pounds of coal has, perhaps, never been excelled.

Triple Expansion Engines. — *Cylinder Proportions for Triple Expansion Engines:* Trans. A.S.M.E., 21–1002, 10–576. *Economy of Triple Expansion Engines:* Trans. A.S.M.E., 8–496.

206. Effects of Condensing. — The effect of the condenser upon the power and economy of engines is indicated in Table 67. The curves in Figs. 215 and 216 were plotted from tests made by Professor R. L. Weighton on a 7, $10\frac{1}{2}$, $15\frac{1}{2} \times 18$ triple-expansion engine at Durham College of Science, Newcastle-on-Tyne. The straight line shows how the mean effective pressure would vary with the degree of vacuum if the power increased directly with the reduction in back pressure. The curved line shows the actual m.e.p., which increases almost along the theoretical line up to a 10-inch vacuum, from which point on the increase is less marked. At 26 inches the actual m.e.p. reaches an apparent maximum. These figures are not applicable to all engines but give a good idea of the limitation of the vacuum with the average type of reciprocating engine with restricted exhaust port openings. With specially designed ports and passages of large cross-sectional area the piston engine shows increase in steam economy up to the highest vacuum carried in the condenser. (See Power, Jan. 16, 1912, p. 72.)

The gain in steam consumption due to the condenser does not indicate a corresponding gain in heat consumption. For example, Engine No. 2, Table 67, shows an apparent gain in steam consumption, due to condensing, of 12.5 per cent, the temperature of the feed water returned to the boiler being 120 degrees F. With a suitable heater the exhaust of the non-condensing engine would be capable of heating the feed water to 210 degrees F. The non-condensing engine should therefore be credited with 210 − 120 or 90 heat units per pound of steam used, or, in round numbers, 9 per cent. The difference between 12.5 per cent and 9 per cent, or 3.5 per cent, represents the net gain in favor of condensing, provided the power necessary to create the vacuum is ignored. Actually, the steam consumption of the condenser pumps might be equal to or greater than 3.5 per cent of the steam generated and the net gain becomes zero or even negative. Referring to Fig. 217, plotted from tests of the 7, $10\frac{1}{2}$, $15\frac{1}{2} \times 18$ triple-expansion engine mentioned above, the solid lines show the feed-water consumption per i.h.p. hour and the broken line the heat units consumed per brake horse power per minute measured above the hot-well temperature. The engine efficiency, based upon the water consumption, increases as the vacuum

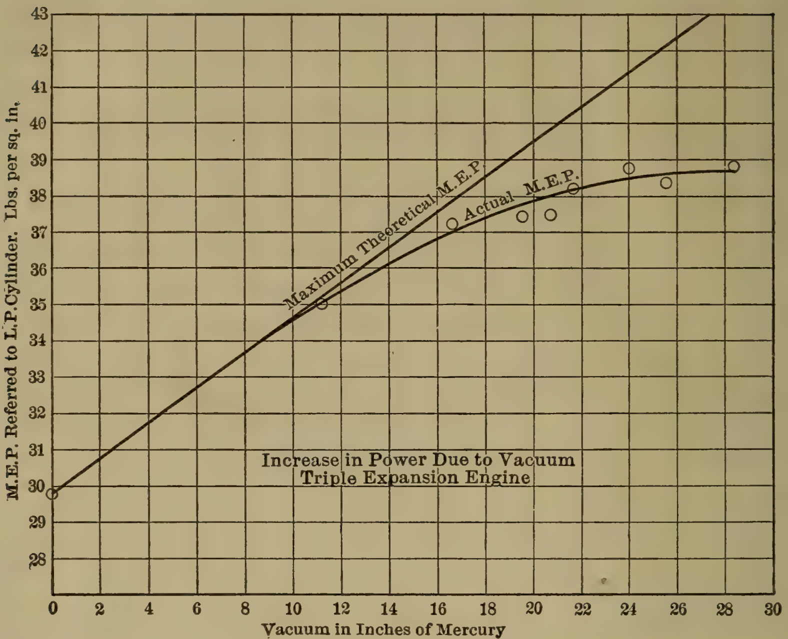


FIG. 215.

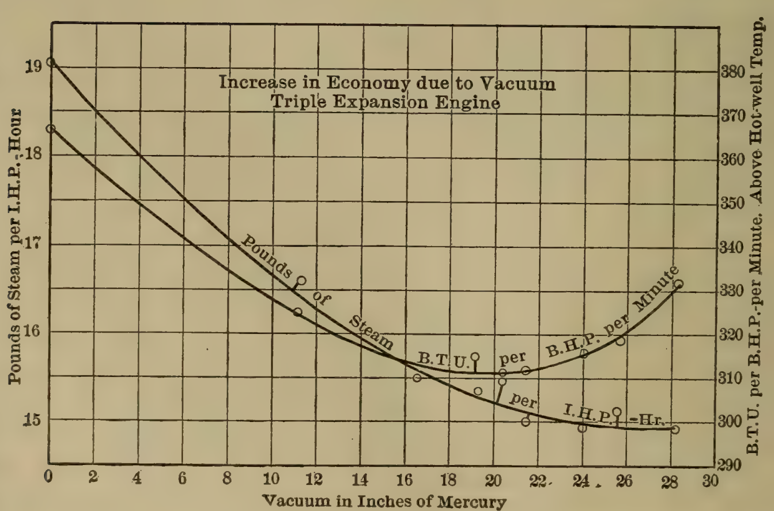


FIG. 216.

increases, reaching a maximum between 26 and 28 inches, whereas the heat-unit curve gives the maximum between 20 and 21 inches. Between 22 and 28 inches the heat-unit curve shows a rapid falling off in economy. Tests of the 5500-horse-power engine at the New York Edison Company's Waterside Station showed that increasing the vacuum from 25.3 to 27.3 inches decreased the water rate only 0.06 pound per

TABLE 67.

EXAMPLES OF THE EFFECT OF CONDENSING ON THE ECONOMY OF RECIPROCATING ENGINES.

	Non-Condensing.			Condensing.				Increase Due to Condensing.	
Reference Number.	Initial Gauge Pressure.	Horse Power Developed.	Steam Consumption, Pounds per H.-P. Hour.	Initial Gauge Pressure.	Back Pressure, Pounds per Square Inch Abs.	Horse Power Developed.	Steam Consumption, Pounds per H.-P. Hour.	In Power, per Cent.	In Economy, per Cent.
1	147	54.7	19.2	149	1.6	83.4	14.8	52.5	25
2	148	540	19.3	147	4	......	16.9	*	12.5
3	126	83	23.8	130	7.4	116	19.1	39.8	19.7
4	67.6	209	28.9	67	4.5	213	22	1.9	23.5
5	103.8	177.5	22.1	103.8	1.2	155	16.5	*	25.1
6	114	160	31	114	......	168	27	2	12.9
7	96	120	23.9	96	4	145	19.4	20.8	18.8
8	118	267	23.24	119	4.2	276.9	16	3.7	31
9	75.9	310	25.6	79	6.4	336	20.5	8.7	19.9
10	62.5	451	30.1	63.6	7.8	444	23	*	23.6
11	186.7	40.4	18.7	184.6	1.6	29.8	12.7	*	32

* Cut-off changed for best economy.

1. 7, 10½, 15½ x 18 triple; Eng. News, Aug. 21, 1902, p. 127.
2. 17, 27 x 24 Westinghouse marine, non-condensing; Power, August, 1903.
3. 1, 18 x 10 Buffalo tandem compound; Elec. World, Sept. 10, 1904, p. 404.
4. 18 x 30 four-valve (slide); Engine Tests, Barrus, p. 88.
5. 21, 65 x 43.31 Corliss; Peabody's Thermodynamics, p. 382.
6. 12 x 12 Reeves simple; Elec. World, Oct. 1, 1904, p. 587.
7. 18 x 48 simple Corliss; Peabody's Thermodynamics, p. 354.
8. 14, 28 x 24 two-valve (slide); Engine Tests, Barrus, p. 175.
9. 17 x 24 two-valve; Engine Tests, Barrus, p. 70.
10. 28 x 36 Corliss; Engine Tests, Barrus, p. 97.
11. Willans triple expansion central valve engine; Peabody, Thermodynamics, p. 406.

i.h.p. hour. (Power, July, 1904, p. 424.) The results are illustrated in Fig. 217. In most cases, and particularly with large compound engines, the net gain due to condensing is considerable, but the feed-water temperatures and power consumed by the auxiliaries should be taken into account.* Fig. 206 shows the effect of vacuum on the steam consumption of a small high-speed simple engine, and Fig. 212 of a cross-compound Corliss. (See also paragraph 236.)

* See Power, Feb. 23, 1909, p. 381.

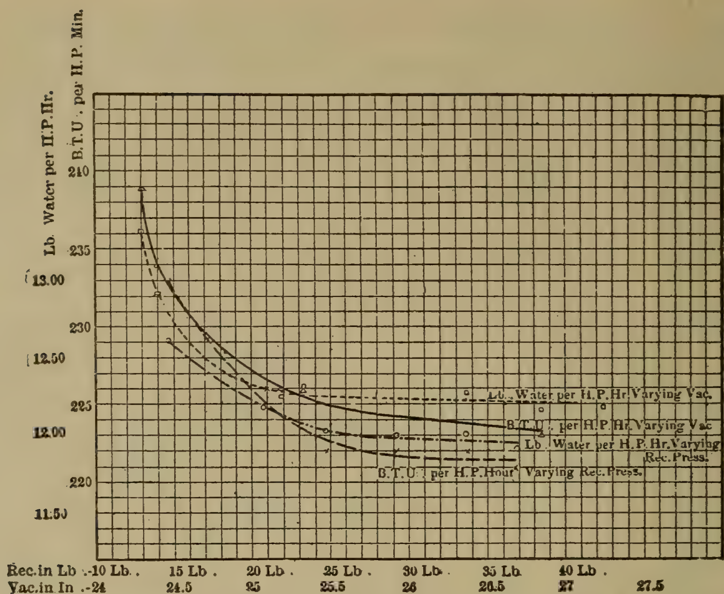


FIG. 217. Performance of 5500-H.P. Engine at Waterside Station of New York Edison Company.

207. Throttling vs. Automatic Cut-Off. — The action of the governor in the throttling engine is shown by the superposed indicator cards (Fig. 218) taken between zero or friction load and maximum load. The effect of throttling is to reduce the pressure during admission, but does not change the point of cut-off or other events of the

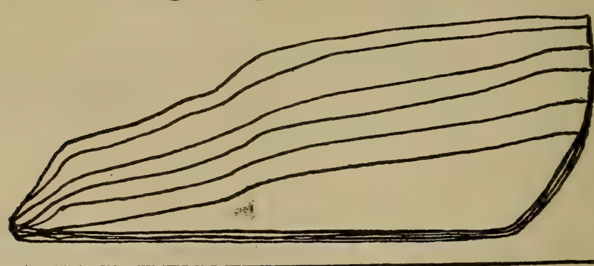


FIG. 218. Typical Indicator Cards. High-speed Throttling Engine.

stroke. The steam may be partially dried or even superheated by throttling, thus tending to reduce cylinder condensation. Initially dry saturated steam at a pressure of 125 pounds gauge would be superheated about 12 degrees in expanding through a throttle to 90 pounds, or if it contained initially 2 per cent moisture would be perfectly dried in expanding to 40 pounds. (See Table 68.) Friction through the valve also tends to dry the steam. Thus with very light loads the

superheat may be decidedly appreciable. The possible gain due to decreased cylinder condensation is to some extent offset by incomplete expansion. The best efficiency for a given load is realized by a proper compromise between cut-off and initial pressure. Experiments made by Professor Denton (Trans. A.S.M.E., 2–150) on a 17 × 30 non-condensing double-valve engine showed the most economical results with $\frac{1}{4}$ cut-off for 90 pounds pressure, $\frac{1}{3}$ cut-off for 60 pounds, and $\frac{45}{100}$ for 30 pounds. The average throttling engine does not give close regulation, the governor usually lacking sensitiveness. Tests show the economy to be better than that of the automatic engine on light loads, and the crank effort more uniform.

TABLE 68.

SHOWING THE INITIAL PER CENT OF MOISTURE THAT WILL BE EVAPORATED IN THROTTLING FROM A HIGHER TO A LOWER PRESSURE.

Based on Marks' and Davis' Steam Tables.

Final Pressures.	Initial Pressure, Absolute.								
	80	85	90	95	100	105	110	115	120
80	......	0.13	0.24	0.36	0.45	0.55	0.65	0.74	0.83
75	0.14	0.26	0.37	0.49	0.59	0.70	0.79	0.88	0.97
70	0.28	0.40	0.52	0.64	0.74	0.84	0.93	1.03	1.12
65	0.43	0.55	0.66	0.78	0.88	0.99	1.08	1.18	1.26
60	0.59	0.71	0.83	0.95	1.06	1.16	1.25	1.34	1.44
55	0.77	0.89	1.01	1.13	1.23	1.34	1.44	1.53	1.62
50	0.97	1.09	1.21	1.33	1.43	1.54	1.64	1.74	1.82
45	1.19	1.32	1.44	1.56	1.66	1.76	1.86	1.96	2.05
40	1.44	1.56	1.68	1.80	1.91	2.02	2.12	2.21	2.30
35	1.72	1.85	1.97	2.10	2.20	2.31	2.41	2.51	2.60
30	2.05	2.18	2.30	2.42	2.53	2.64	2.74	2.84	2.93
25	2.44	2.56	2.69	2.82	2.92	3.03	3.13	3.23	3.32
20	2.90	3.04	3.16	3.29	3.40	3.51	3.61	3.71	3.80
15	3.51	3.65	3.78	3.90	4.01	4.13	4.23	4.33	4.43

Final Pressures.	Initial Pressure, Absolute.								
	125	130	135	140	145	150	155	160	165
80	0.91	0.99	1.08	1.15	1.22	1.29	1.35	1.41	1.48
75	1.05	1.13	1.21	1.28	1.36	1.43	1.49	1.55	1.62
70	1.19	1.27	1.36	1.43	1.50	1.58	1.64	1.70	1.77
65	1.34	1.43	1.51	1.59	1.66	1.73	1.79	1.85	1.93
60	1.52	1.60	1.68	1.76	1.83	1.90	1.96	2.03	2.10
55	1.70	1.78	1.86	1.94	2.02	2.09	2.15	2.21	2.29
50	1.90	1.99	2.08	2.15	2.22	2.30	2.36	2.42	2.50
45	2.13	2.21	2.30	2.38	2.45	2.52	2.59	2.65	2.73
40	2.39	2.47	2.55	2.63	2.71	2.78	2.84	2.91	2.99
35	2.68	2.77	2.85	2.93	3.01	3.08	3.14	3.21	3.29
30	3.01	3.10	3.18	3.26	3.34	3.41	3.48	3.55	3.63
25	3.41	3.49	3.58	3.66	3.74	3.81	3.88	3.96	4.04
20	3.88	3.97	4.06	4.15	4.22	4.30	4.37	4.45	4.53
15	4.51	4.60	4.70	4.78	4.86	4.94	5.01	5.09	5.17

The indicator cards shown in Fig. 219 were taken from a single-valve high-speed automatic engine operating between friction load and maximum load. The mean effective pressure is adjusted to suit the load by the automatic variation in the cut-off, the initial pressure remaining the same. Since the cut-off is controlled by the action of the governor on the single valve, all other events of the stroke are likewise changed. With a four-valve engine the variation in cut-off does not affect the other events.

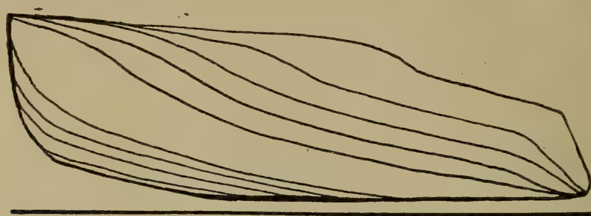


FIG. 219. Typical Indicator Cards. High-speed Automatic Engine.

The chief advantage of the automatic over the throttling engine lies in its sensitive regulation, and while, in general, it gives a lower steam consumption than the throttling engine, this is probably in most cases due to superior construction and not to the method of governing.

The following performances of a Belliss 250-horse-power high-speed condensing engine fitted with both automatic and throttling governing devices give results decidedly in favor of the throttling engine. (Pro. Inst. of Mech. Engrs., 1897, p. 331.)

	Automatic Cut-Off.				Throttling.			
Percentage of load........	100	62.5	33	25	100	62.5	33	25
Electrical horse power....	213	132	77.8	53	213	132	77.8	53
Steam per I.H.P. hour....	22.5	22.9	28.5	34.3	21	21.7	25.6	28.4

Some of the comparative advantages and disadvantages of the automatic and throttling engines are as follows:

AUTOMATIC.	THROTTLING.
Advantages.	
1. Sensitiveness of regulation.	1. Low first cost.
2. Increased ratio of expansion.	2. Crank effort more uniform.
3. Low terminal pressures.	3. Reduced cylinder condensation.
	4. Simplicity of regulating device.
Disadvantages.	
1. Increased cylinder condensation.	1. Low ratio of expansion.
2. Greater variation in crank effort.	2. High terminal pressure.
3. Complicated valve gear.	3. Low initial pressure at early loads.
4. Low economy at very early loads.	

Fig. 220 shows the relative steam consumptions of an engine under the same conditions of load when controlled by variable expansion and by throttling. Suppose this engine to be altered in capacity so that the m.e.p. referred to the low-pressure piston is about 32, then the steam consumption with the throttling governor will be as shown by straight line A. This shows that between 32 and 12 pounds m.e.p. very little is gained by a variable expansion, and below that load the throttled governor is the more economical. (Power, Feb. 21, 1911, p. 301.)

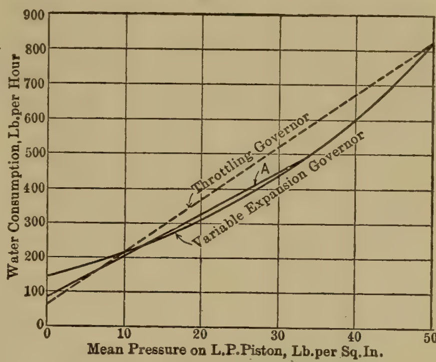
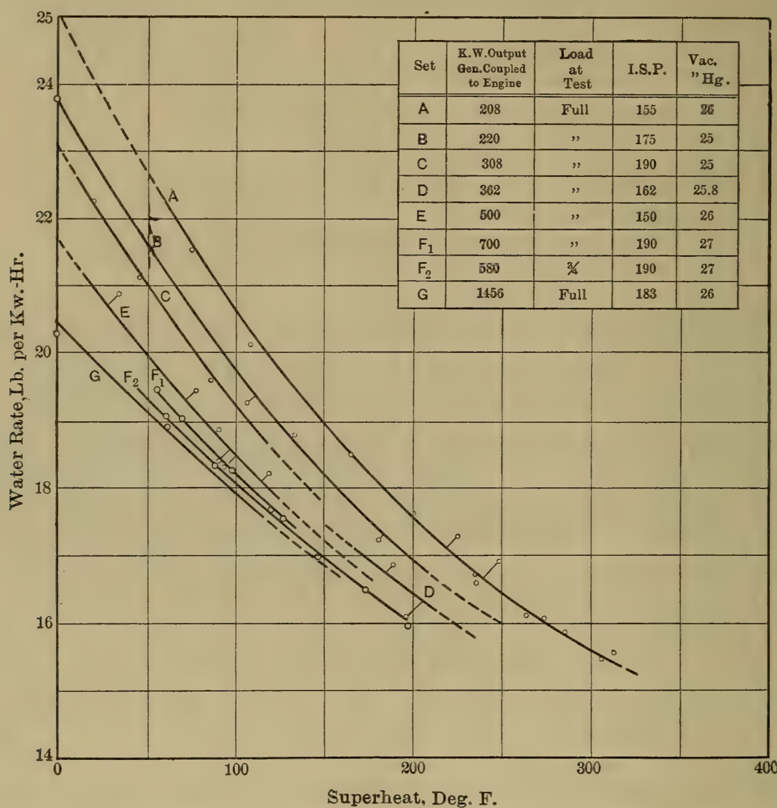


Fig. 220. Throttling *vs.* Automatic Cut-off.

208. Influence of Superheat. — (See also paragraph 117.) Table 69 gives test results for several different types of engine employing superheated steam. These figures may be compared with the performances of engines using saturated steam as given in Tables 59 and 66. A decided gain in economy is shown in favor of superheat for single-cylinder engines. With compound engines the advantage is not so apparent, while triple-expansion engines show the least gain. Tables 69 to 72 show the effect of superheating on simple, compound and triple-expansion engines and represent average current practice. (Proc. A.S.M.E., September, 1907.) Better results than these have been obtained with such engines as the Lentz compound, but for ordinary superheated steam practice the results in the tables may be used with confidence. Some idea of the wonderful fuel economy effected in Europe with the use of highly superheated steam in connection with the so-called *locomobile* is gained from the results shown in Table 73. This type of engine has not yet been introduced to any extent in this country but it is only a matter of time when the cost of coal will advance to such a point as to preclude all but the more economical types of prime movers.

As far as steam consumption is concerned, all engines show greater economy with superheated than with saturated steam, but the thermal gain is not so marked, and when the economy is measured in dollars and cents per developed horse power, taking all things into consideration the gain is still further reduced and in some cases completely neutralized. First cost, maintenance, and disposition of the exhaust must all be considered in determining the ultimate commercial gain due to the use of superheated steam.

Fig. 221 gives the results of a series of tests made on a number of Belliss & Morcom engines using superheated steam. (Pro. Inst. of Mech. Engrs., March, 1905, p. 302.) The engines were from 200 to 1500 kilowatts capacity and were tested at full load. It is noticeable that the curves all converge to a single point and will meet at about 400 degrees F. The results show that if sufficient superheat is put into

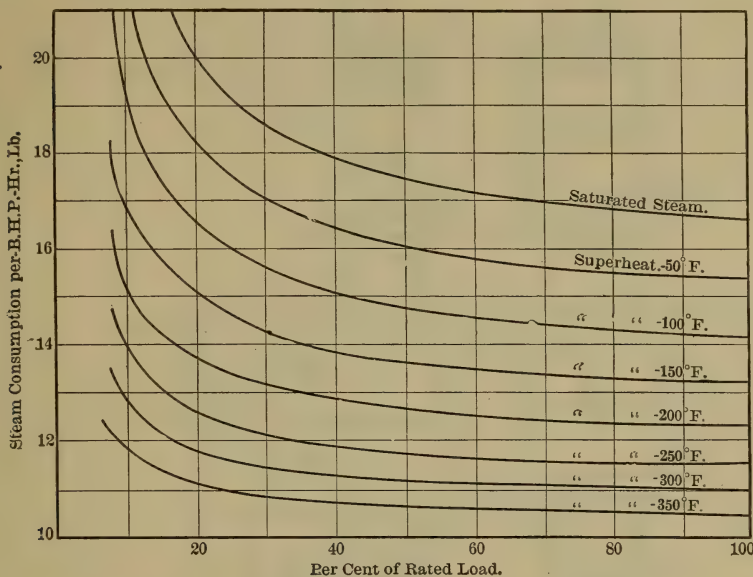


Set	K.W. Output Gen. Coupled to Engine	Load at Test	I.S.P.	Vac. "Hg.
A	208	Full	155	26
B	220	"	175	25
C	308	"	190	25
D	362	"	162	25.8
E	500	"	150	26
F_1	700	"	190	27
F_2	580	¾	190	27
G	1456	Full	183	26

FIG. 221. Effect of Superheat on Steam Consumption.

the steam all engines of whatever size are equally economical. Fig. 222 shows the relationship between degree of superheat and the heat consumption at various loads for a 300-horse-power Belliss & Morcom high-speed triple-expansion engine. (Pro. Inst. Mech. Engrs., March, 1905, p. 303.) It will be noted that the variation in heat consumption at different percentages of load becomes less marked as the degree of superheat increases. With superheat of 350 degrees F. the heat consumption from $\frac{1}{4}$ load to full load is practically constant.

These curves though strictly applicable to the specific cases cited are more or less general and represent the influence of superheat on all types of piston engines.



.FIG. 222. Effect of Superheat on Steam Consumption.

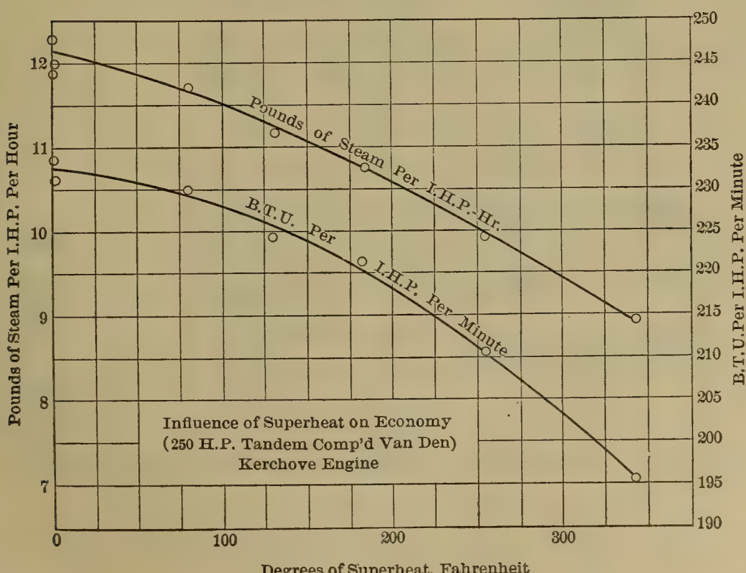


FIG. 223.

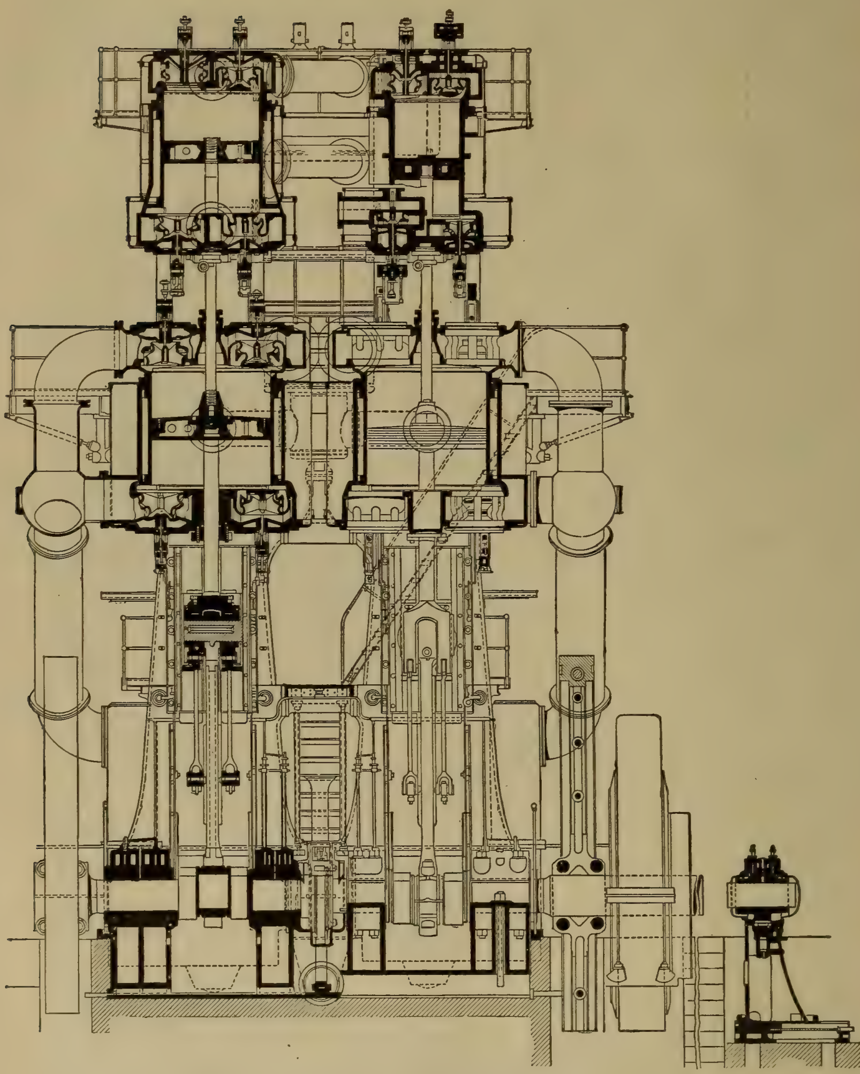


FIG. 224. 3000 H.P. Sulzer Engine Designed for Highly Superheated Steam.

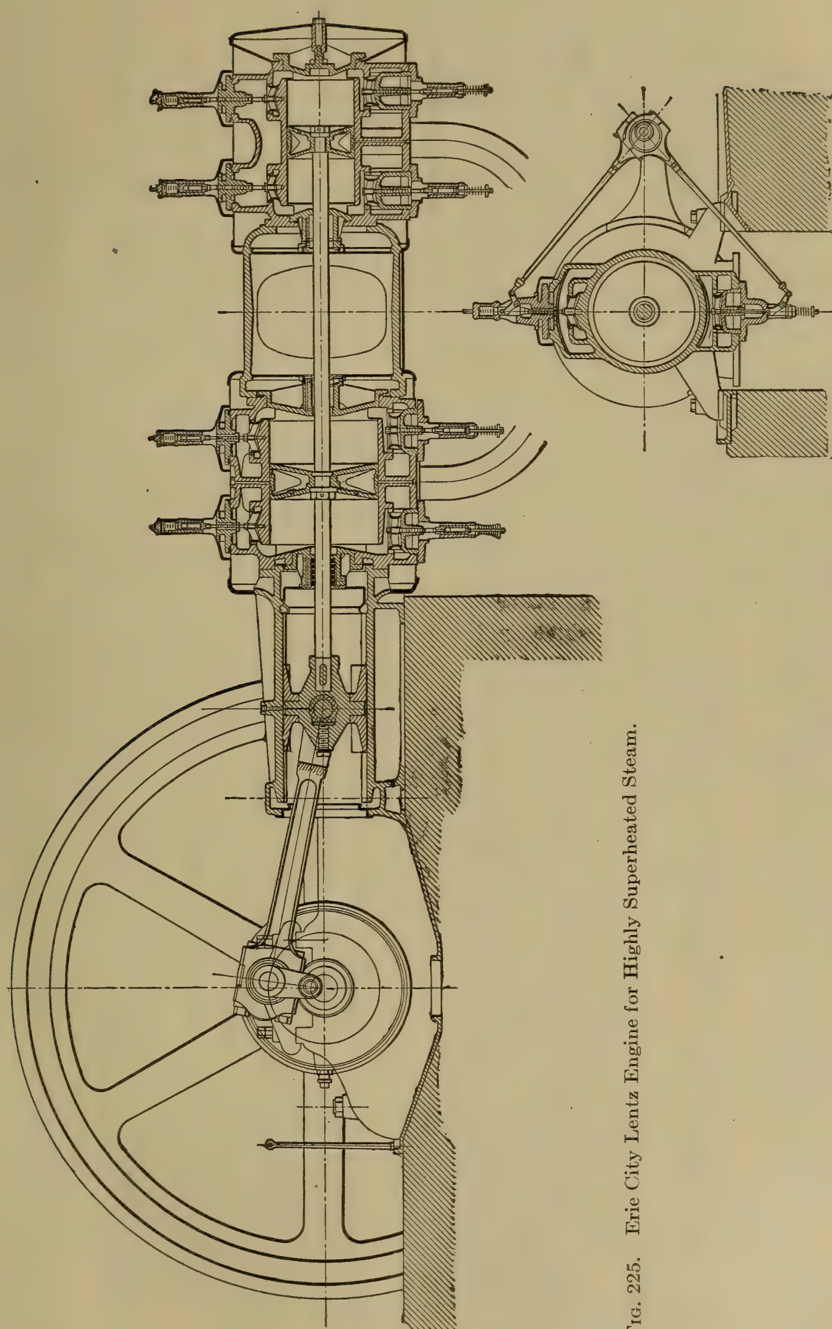


Fig. 225. Erie City Lentz Engine for Highly Superheated Steam.

TABLE 69. — EXAMPLES OF STEAM-ENGINE ECONOMY.

SUPERHEATED STEAM.

Index.	Kind of Engine.	References.	Cylinder Dimensions, Inches.	Cylinder Ratio.	Horse Power.	Initial Press., Lbs. Gauge.	Back Press., Inches Mercury.	R.P.M.	Temp. Hot Well, Deg. F. *	Lbs. of Steam per I.H.P. Hr.	B.T.U. per I.H.P. per Min. ‡	Thermal Eff., per Cent.	B.T.U. per I.H.P., Min., Perfect Eng.	Efficiency Ratio, per Cent.	Superheat at Admission, Deg. F.	Temp. Steam at Admission, Deg. F.	Spec. Heat of the Superheated Steam.
	Triple Expansion.																
1	Binary Vapor Eng., Royal High School, Berlin.	Jour. Franklin Inst., Dec., 1902, p. 456.			211	143	4.5	143.5		8.60	158.3	26.8			221	590	
2	Sulzer, Four-Cylinder....	Eng. News, Oct. 2, 1902, p. 259.	32, 47, 58 x 59	1:2.03:3.27	2860	173	2.0	85	102	8.97	187.7	22.6	156	73.5	230	606	0.54
3	Sulzer, Three-Cylinder...	Zeit. d. V.D.I., Aug., 1905, p. 1353.	15.5,25.4,37.5 x 25.6	1:2.64:5.9	549	166	2.4	144.4	108	10.00	207.3	20.4	142	68.5	229.7	602	0.54
4	Sulzer, Three-Cylinder...	Engr., Lond., May 25, 1900, p. 546.	34, 46, 61 x 51	1:2.08:3.22	2940	167	1.6	82.5	95	9.58	204.0	20.8	136	66.7	264	637	0.53
5	Worthington Pumping Eng., Central Pk., Chic.	Eng. News, May 26, 1904, p. 487.			646	146.8	1.6	18.6	95	10.00	196.6	21.6	142	72.5	87.2	451.1	0.57
6	Riedler Pumping Engine, Chicago Ave. Sta., Chic.	Engr. U.S., Nov. 15, 1907, p. 1092.	15, 29, 48 x 48		590	170	2.6	62	111	9.73	196.5	21.6	136	69.3	166.3	541.9	0.55
	Compound.																
7	Cole, Marchent & Morley, Cross-Comp., Jacketed.	Engr., London, June, 1905, p. 546.	21, 36 x 36.	1:2.94	145.5	114.5	1.72	100.7	97	8.58	176.1	24.0	143	82.0	202	548	0.52
8	Van den Kerchove, Tandem, Heads Jacketed.	Amer. Elecn., May, 1903, p. 217..	12.8, 22 x 33.4	1:2.97	212	131	2.2	127	105	8.99	194.8	21.7	142	73.0	342	699	0.51
9	Van den Kerchove, Tandem, Heads Jacketed.	Amer. Elecn., May, 1903, p. 217..	12.8, 22 x 33.4	1:2.97	217	129.5	2.2	127	105	10.75	218.2	19.4	151	69.1	183	539	0.54
10	Easton & Co., Tandem-Compound.	Amer. Elecn., April, 1903, p. 178.	15, 24 x 28	1:2.67	239	120	2.0	140	102	9.00	187.0	22.6	147	78.5	240	590	0.52
11	Rice & Sargent, Melbourne Mills, Pa.	Trans. A.S.M.E., Vol. 25, p. 278.	16, 28 x 42	1:3.06	920	142	4.0	102	126	9.56	188.3	22.5	136	72.2	296	658	0.52
12	McIntosh & Seymour, Edison Co., So. Boston.	Trans. A.S.M.E., Vol. 25, p. 491.	29, 60 x 56	1:4.3	2202	162.8	4.4	98.2	130	11.57	222.6	19.1	123	54.3	98.4	470	0.58
13	Cross-Compound, Cylinders Jacketed.	Barrus, Eng. Tests, p. 202........	18, 48 x 48	1:7.3	659	143	3.4	80	120	11.89	223.5	19.0	123	55.2	40	402	0.59
14	Sulzer, Tandem-Comp. ..	Eng. News, Oct. 2, 1902, p. 259. ..	26.8, 47.2 x 67	1:3.1	788	116	2.8	65	114	9.68	207.2	20.3	148	71.5	343	690	0.51
15	White, Auto. Engine....	Trans. A.S.M.E., Vol. 28.........	3, 6 x 4.5	1:4.0	40	426	Atmos.	850	212	11.96	244.0	17.4	166	68.0	316	766	0.58
	Simple.																
16	Poppet-Valve, Condens'g.	Zeit. d. V.D.I., Aug., 1905, p. 1310.	16.3 x 39.4		123	145	1.4	81.1	91	16.70	326.2	13.0	146	46.0	73.8	424	0.57
17	Poppet-Valve, Condens'g.	Zeit. d. V.D.I., Aug., 1905, p. 1310.	16.3 x 39.4		120	145	1.5	81.2	92	14.70	307.0	13.8	146	47.6	226.2	576	0.53
18	Poppet-Valve, Non-Condensing.	Zeit. d. V.D.I., Aug., 1905, p. 1310.	16.3 x 39.4		123	145	Atmos.	81.5	212	16.10	307.5	13.8	242.2	79.0	254.3	604.3	0.52

* Ideal feed-water temp. ‡ Above ideal feed-water temp.

TABLE 70.

STEAM AND COAL SAVING IN A SIMPLE NON-CONDENSING ENGINE OF 250 I.H.P. WITH SUPERHEATED STEAM AT DIFFERENT TEMPERATURES.

Press. 12 atm.=177 lb.; temp. of sat. steam 369 deg. F.; cut-off 20 per cent; piston speed 10 ft. per sec.; slide or piston valve; change of cut-off effected by valve gear.

Kind of Superheat.	Degrees of Superheat.	Temperature of Steam.	Cut-off Constant. I.H.P. Variable.							Cut-Off Variable. I.H.P. Constant.								Remarks.
			I.H.P.	At Engine.			At Boiler.			I.H.P.	At Engine.			At Boiler.			Cut-off, per Cent.	
				Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat Steam.		Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat. Steam.			Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat. Steam.		Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat. Steam.			
					Steam, per Cent.	Fuel, per Cent.		Feed Water, per Cent.	Fuel, per Cent.			Steam, per Cent.	Fuel, per Cent.		Feed Water, per Cent.	Fuel, per Cent.		
	Deg. F.	Deg. F.																
None..........	0	369	250	27.00	......	......	29.00	......	....	250	27.00	......	......	29.00	......	......	20.00	Indirect; superheater in boiler
75 to 150° F. Low:																		
Indirect.....	103	472	235	21.00	22.0	18.5	21.40	27	23	250	21.05	22.0	18.5	21.47	26.0	22.5	20.88	Direct; superheater separately fired
Direct.......	103	472	235	21.00	22.0	10.0	21.40	27	14	250	21.05	22.0	9.5	21.47	26.0	14.0	20.85	
150 to 225° F. Medium:																		
Indirect.....	162	531	230	19.85	26.5	20.5	20.10	30	27	250	20.00	25.5	20.0	20.40	29.5	24.0	21.35	
Direct.......	162	531	230	19.85	26.5	21.5	20.10	30	19	250	20.00	25.5	11.0	20.40	29.5	15.5	21.35	
225 to 290° F. High:																		
Indirect.....	234	603	222	18.50	31.0	23.0	18.85	35	31	250	18.75	30.5	22.0	19.13	34.0	26.0	21.75	
Direct.......	234	603	222	18.50	31.0	15.0	18.85	35	22	250	18.75	30.5	13.5	19.13	34.0	18.0	21.75	

TABLE 71.

STEAM AND COAL SAVING IN A COMPOUND ENGINE, CONDENSING, OF 250 I.H.P. WITH SUPERHEATED STEAM AT DIFFERENT TEMPERATURES.

Press. 10 atm. = 142.23 lbs.; temp. of sat. steam 354 deg. F.; cut-off 6 per cent; piston speed 10 ft. per sec.; automatic cut-off; 4 poppet or piston valves per cyl.

Kind of Superheat.	Degrees of Superheat.	Temperature of Steam.	Cut-off Constant. I.H.P. Variable.							Cut-off Variable. I.H.P. Constant.								Remarks.
			I.H.P.	At Engine.			At Boiler.			I.H.P.	Variation of Cut-off, per Cent.	At Engine.			At Boiler.			
				Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat. Steam. Steam, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	Feed Water per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Feed Water, per Cent.	Saving over Sat. Steam. Fuel, per Cent.			Steam Consump. per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Steam, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	Feed Water, per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Feed Water, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	
	Deg. F.	Deg. F.																
None	0	354	250	14.72	......	......	15.73	......	....	250	6.00	14.72	......	......	15.73	......	......	Indirect; superheater in boiler
75 to 150 deg. F.																		
Low: Indirect	130	484	225	12.25	15.0	11.0	12.50	21	17	250	6.95	12.50	15.0	11.5	12.70	20.0	16.0	Direct; superheater separately fired
Direct	130	484	225	12.25	15.0	1.5	12.50	21	8	250	6.95	12.50	15.0	1.5	12.70	20.0	7.0	
150 to 225 deg. F.																		
Medium: Indirect	202	556	215	11.60	21.0	15.0	11.81	25	21	250	7.50	11.75	20.0	14.0	11.98	24.5	18.5	
Direct	202	556	215	11.60	21.0	5.5	11.81	25	12	250	7.50	11.75	20.0	4.5	11.98	24.5	9.5	
Double: (1) Indirect. (2) Direct	202	556	205	10.70	27.5	18.0	10.93	31	24	250	8.00	10.85	26.0	17.0	11.09	30.0	21.0	
(1) Direct (2) Indirect.	202	556	205	10.70	27.5	16.5	10.93	31	23	250	8.00	10.85	26.0	15.0	11.09	30.0	19.5	
225 to 290 deg. F.																		
High: Indirect	274	628	205	10.70	27.5	19.0	10.93	31	27	250	8.00	10.85	26.0	17.5	11.09	30.0	21.5	
Direct	274	628	205	10.70	27.5	10.0	10.93	31	17	250	8.00	10.85	25.0	8.5	11.09	30.0	13.0	
Double: (1) Indirect (2) Direct	274	628	198	10.27	30.0	18.5	10.50	34	26	250	8.50	10.51	28.5	16.0	10.75	32.0	20.5	
(1) Direct (2) Indirect	274	628	198	10.27	30.0	17.0	10.50	34	25	250	8.50	10.51	28.5	15.0	10.75	32.0	19.0	
290 to 360 deg. F.																		
Very high: Indirect	338	692	198	10.27	30.0	20.5	10.50	34	28	250	8.50	10.51	28.5	18.5	10.75	32.0	22.5	
Direct	338	692	198	10.27	30.0	11.5	10.50	34	20	250	8.50	10.51	28.5	9.0	10.75	32.0	13.5	

TABLE 72.

STEAM AND COAL SAVING IN A TRIPLE EXPANSION ENGINE, CONDENSING, OF 5000 I.H.P. WITH SUPERHEATED STEAM AT DIFFERENT TEMPERATURES.

Press. 14 atm. = 199 lb.; temp. of sat. steam 381 deg. F.; cut-off 4 per cent; piston speed 16.5 ft. per sec.; automatic cut-off; 4 poppet or piston valves per cyl.

Kind of Superheat.	Degrees of Superheat.	Temperature of Steam.	Cut-off Constant. I.H.P. Variable.							Cut-off Variable. I.H.P. Constant.								Remarks.
			I.H.P.	At Engine.			At Boiler.			I.H.P.	Variation of Cut-off, per Cent.	At Engine.			At Boiler.			
				Steam Consump. per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Steam, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	Feed Water, per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Feed Water, per Cent.	Saving over Sat. Steam. Fuel, per Cent.			Steam Consump. per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Steam, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	Feed Water per I.H.P. Hour, Lbs.	Saving over Sat. Steam. Feed Water, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	
	Deg. F.	Deg. F.																
None	0	381	5000	11.82	. . . . . .	. . . . . .	12.70	. . . . . .	. . . .	5000	4.00	11.82	. . . . . .	. . . . . .	12.70	. . . . . .	. . . . . .	Indirect; superheater in boiler
75 to 150 deg. F.																		
Low: Indirect	144	525	4350	10.50	11.5	7.5	10.70	16	12	5000	4.85	10.90	7	3.0	11.25	11.5	7.5	Direct; superheater separately fired
Direct.	144	525	4350	10.50	11.5	−3.0	10.70	16	2	5000	4.85	10.90	7	−8.0	11.25	11.5	−2.5	
150 to 225 deg. F.																		
Medium: Indirect . .	216	597	4150	9.81	17.0	10.0	10.03	21	17	5000	5.15	10.32	13	6.0	10.52	17.5	11.0	
Direct.	216	597	4150	9.81	17.0	0.0	10.03	21	8	5000	5.15	10.32	13	−4.5	10.52	17.5	1.0	
Double: (1) Indirect																		
(2) Direct .	216	597	4000	9.15	22.5	12.5	9.35	26	19	5000	5.40	9.60	19	8.5	9.80	23.0	13.5	
(1) Direct .																		
(2) Indirect	216	597	4000	9.15	22.5	11.0	9.35	26	17	5000	5.40	9.60	19	6.5	9.80	23.0	12.0	
225 to 290 deg. F.																		
High: Indirect	288	669	4000	9.15	22.5	13.0	9.35	26	21	5000	5.40	9.60	19	9.0	9.80	23.0	14.0	
Direct.	288	669	4000	9.15	22.5	4.0	9.35	26	12	5000	5.40	9.60	19	−1.0	9.80	23.0	5.0	
Double: (1) Indirect																		
(2) Direct .	288	669	3775	8.48	28.0	16.0	8.70	31	24	5000	5.85	9.20	22	9.0	9.40	26.0	14.5	
(1) Direct .																		
(2) Indirect	288	669	3775	8.48	28.0	14.5	8.70	31	22	5000	5.85	9.20	22	7.5	9.40	26.0	13.0	
290 to 360 deg. F.																		
Very high: Indirect	353	734	3775	8.48	28.0	18.0	8.70	31	25	5000	5.85	9.20	22	11.5	9.40	26.0	16.5	
Direct. . .	353	734	3775	8.48	28.0	8.0	8.70	31	16	5000	5.85	9.20	22	1.5	9.40	26.0	7.0	

209. The Locomobile. — Fig. 226 shows a section through the engine cylinders and the boiler setting of a Wolf tandem-compound *locomobile,* illustrating a type of steam plant which is finding much favor in Europe. The entire equipment is self-contained and requires very little floor space. The engine is set upon the boilers with the cylinders projecting into the smoke box so as to minimize piping and radiation losses. Steam is generated at a pressure of 175–225 pounds absolute and is superheated by the furnace gases to 800–850 degrees F. before it is admitted to the high-pressure cylinder. Exhaust steam from the high-pressure cylinder

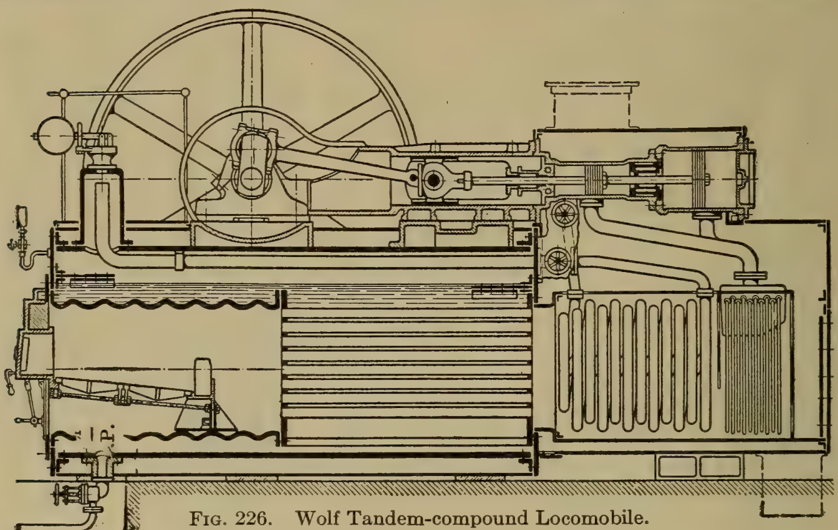


FIG. 226. Wolf Tandem-compound Locomobile.

is reheated by an auxiliary superheater (adjoining the main superheater) to a temperature of 450–550 degrees F. before it enters the low-pressure cylinder. The feed water is heated by an economizer placed in the breeching. All auxiliaries are driven by the main engine.

Locomobiles are made in various sizes ranging from 25 to 1000 horse power and are designed for condensing or non-condensing operation. Coal consumptions as low as 0.8 pound per developed horse-power hour have been realized and one pound per horse-power hour is common practice. On account of the extremely high degree of superheat employed the load curve is very flat and there is little difference in economy between the small and large units.

Table 73 gives the results of tests of a 100-horse-power Wolf tandem-compound locomobile and illustrates the remarkable economy which is being effected in Europe with this type of plant.

The lowest steam consumption recorded to date, 6.95 pounds per indicated horse-power hour, is credited to an engine of this class. (See Zeit. d. Ver. Deut. Ingr., Mar. 18, 1911, p. 415.)

TABLE 73.

A REMARKABLE ENGINE PERFORMANCE.*

200, 400 × 400 mm. Locomobile.

(7.8, 15.7 × 15.7 in.)

Number of Test.	Initial Pressure, Lbs. per Sq. In. Abs.	Condenser Pressure, Lbs. Abs.	Steam Temperatures, Degrees F.				R.P.M.
			Entering High-pressure Cylinder.	Leaving High-pressure Cylinder.	Entering Low-pressure Cylinder.	Final Feed Water.	
		Condensing with Intermediate Superheating.					
1	220	1.47		Saturated.		242	236
2	227	1.17	712	377	462	212	241
3	220	1.17	718	367	460	206	242
4	221	1.17	806	426	530	221	246
5	220	1.17	842	469	538	...	243
6	220	1.17	872	520	...	241	243
		Non-condensing without Intermediate Superheating.					
7	220	....	832	...	462	289	237
8	220	....	856	...	505	284	238
9	221	....	878	...	527	284	242
10	220	....	869	...	572	257	241
11	220	....	817	...	525	248	241
12	221	....	878	...	568	259	241

* Compiled from Zeit. des Ver. deut. Ingr., June, 1911.

No. of Test.	I.h.p.	D.h.p.	Mechanical Efficiency, Per Cent.	Steam Consumption, Pounds		Coal Burned, Lbs. per D.h.p. Hr.	Heat Consumption, B.t.u. per I.h.p. per Minute.*
				Per I.h.p. Hr.	Per D.h.p. Hr.		
		Condensing with Intermediate Superheating.					
1	112.5	103.2	91.6	13.98	14.19	1.59	260
2	138.4	132.8	96.0	8.51	8.87	1.00	198
3	140.3	131.4	93.5	8.33	8.90	1.00	195
4	140.4	133.4	95.0	7.68	8.06	0.96	186
5	138.8	132.5	95.5	7.24	7.56	0.87	175
6	141.8	134.0	94.5	7.15	7.56	0.86	175
		Non-condensing without Intermediate Superheating.					
7	61.5	49.3	78.0	11.22	14.43	1.65	262
8	83.8	74.0	88.0	10.60	11.84	1.17	249
9	111.0	98.5	88.0	9.95	11.38	1.12	237
10	129.9	120.8	93.0	10.00	10.88	1.07	238
11	140.4	132.2	94.0	10.68	11.34	1.12	248
12	142.1	132.4	93.0	9.93	10.66	1.05	235

* Above ideal feed-water temperature corresponding to exhaust pressure.

210. Uniflow or Straight-flow Engine. — Fig. 227 shows a longitudinal section through a single-cylinder engine designed by Professor J. Stumpf, of Charlottenberg College, Germany, which is finding much favor with the European engineers, over a half-million horse power being in service at this writing. A 300-horse-power engine of this design is credited with a steam consumption of 8.5 pounds per i.h.p. hour; initial pressure 130 pounds absolute; superheat 261 degrees F., a performance equalled only by the best compound and triple-expansion engines. The engine is similar in principle to the two-cycle gas engine in which the elongated piston acts as an exhaust valve, opening and closing a series of slots in the middle of the cylinder shell. The live steam enters the engine through the cylinder head, which it heats, and is admitted into the cylinder through a double-seated poppet valve, and, after expansion, is exhausted through the slots in the middle of

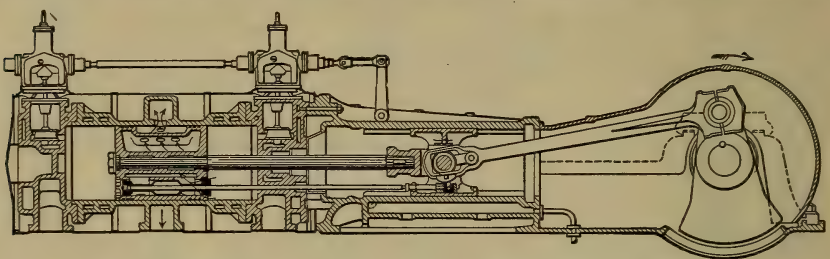


Fig. 227. Longitudinal Section through Stumpf Straight-flow Engine.

the cylinder. A high degree of expansion is possible without excessive cylinder condensation, since the exhaust steam does not come into contact with the live-steam-heated surfaces of the cylinder as in the ordinary type of piston engine.

On account of the high degree of expansion used in this type of engine the cylinders are necessarily large per unit output. The high compression necessitates the use of heavy cylinders and flywheel and the inertia of the reciprocating parts requires a massive foundation. The cost per unit of power is therefore higher than that of the ordinary single-cylinder high-speed engine. The cost is, however, less than that of slow-speed compound and triple-expansion engines with which it successfully competes. Table 74 gives a comparison of the results of triple-expansion engines and a number of uniflow engines.

The uniflow engine has not been generally adopted in this country notwithstanding its excellent showing in Europe.

Fig. 228 shows a section through a uniflow engine as manufactured by the Nordberg Manufacturing Company, Milwaukee, Wis. The curves in Fig. 229 are based upon the tests of a 20 × 30, 200-horse-power

Nordberg uniflow engine using very wet steam. The dotted lines show assumed results within the capacity of the machine.

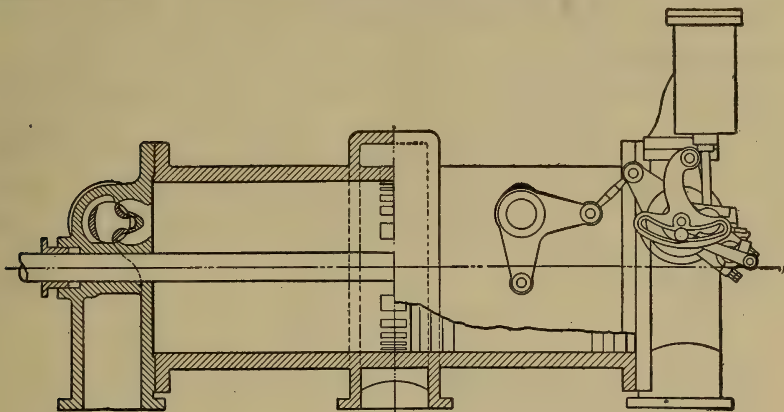


FIG. 228. Section through Cylinder of Nordberg Uniflow Engine.

TABLE 74.

COMPARISON OF OPERATING PERFORMANCES OF TRIPLE-EXPANSION STEAM ENGINES AND STRAIGHT-FLOW STEAM ENGINES.

(Power, Jan. 31, 1911.)

Manufacturers.	Indicated Horse Power.	Diameter of Cylinders in Inches.			Stroke in Inches.	Rev. per Min.	Number of Moving Parts.	Steam.		Steam Consumption per	
		High.	Intermediate.	Low.				Pressure.	Temperature.	I.h.p. per Hour.	Kw. per Hour.
	Best Performance of Stationary Triple-expansion Engines.										
Sulzer, Switzerland	6000	40	60	2×73	67	83	228	170	572	8.5	13.7
Gorlitz, Germany	6000	40	60	2×73	67	83	......	170	572	8.5	13.7
Nurnberg, Germany	6000	41	60	2×73	67	83	......	170	572	8.5	13.7
	Straight-flow Steam Engines.										
Sulzer, Switzerland	300	......	23.5	......	31.5	155	33	130	617	8.5	13.7
Same engine	300	......	23.5	......	31.5	155	33	130	0	10.6	......
Gebr. Stock, Holland	80	......	12.6	......	19.8	200	......	149	662	9.7	......
Burmeister, Denmark	......	......	17.8	......	23.5	180	......	138	662	9.5	......

211. Binary-vapor Engines. — A consideration of the Carnot or Rankine cycles shows that theoretically the efficiency of the steam engine may be increased by raising the temperature of the steam supplied or by lowering the temperature of the exhaust, that is to say, by increasing the range. Superheated steam development has practically determined the upper limit, and economical practice indicates a vacuum of about 26 inches, corresponding to 126 degrees F., as the average lower limit for most efficient results from a commercial standpoint.

In the binary-vapor engine the working range has been considerably increased by substituting a highly volatile liquid, as sulphur dioxide, for the water which is ordinarily used as the cooling medium in the surface condenser.

The SO_2 in condensing the exhaust steam is itself vaporized and the vapor, under a pressure of about 175 pounds per square inch, used expansively in a secondary reciprocating engine. The exhausted SO_2 is discharged into a surface condenser in which it is liquefied by cooling water much the same as in refrigerating practice and used over and over again. Referring to Fig. 230, which illustrates diagrammatically a binary-vapor engine at the Royal Technical High School, Berlin:

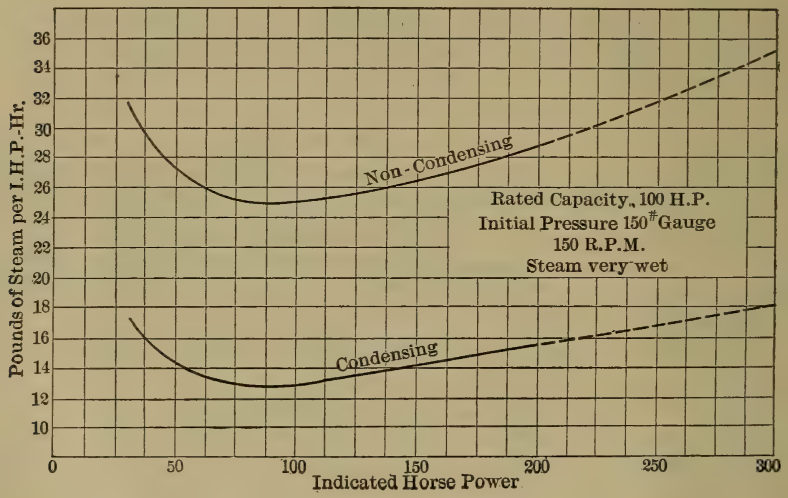


FIG. 229. Performance of Nordberg Uniflow Engine.

A, *B*, and *C* are the three steam cylinders of an ordinary triple-expansion engine and *D* the SO_2 cylinder. All four cylinders drive a common crank shaft *E*. *F* is a high-pressure surface condenser which acts as a vaporizer for the SO_2 and a condenser for the steam. *G* is a surface condenser which serves to condense the SO_2 vapor. *H* is a liquid SO_2 tank. The operation is as follows: Highly superheated steam enters the high-pressure steam cylinder at *I* and leaves the low-pressure cylinder at *J*, just as in any steam engine. The exhaust steam enters chamber *F* and is condensed by the liquid SO_2 passing through the coils. The condensed steam and entrained air are removed from the chamber by a suitable air pump. The steam in condensing gives up its latent heat to the liquid SO_2 and causes it to vaporize. The SO_2 vapor passes from the coils in chamber *F* to the SO_2 engine *D* and

performs work. The exhausted SO_2 vapor flows from cylinder *D* to chamber *G*, and is condensed by cooling water flowing through a series of tubes. The liquid SO_2 is collected in liquid tank *H* and thence is pumped into the coils in vaporizer *F*. The approximate temperatures and pressures at different points of the cycle are indicated on the diagram.

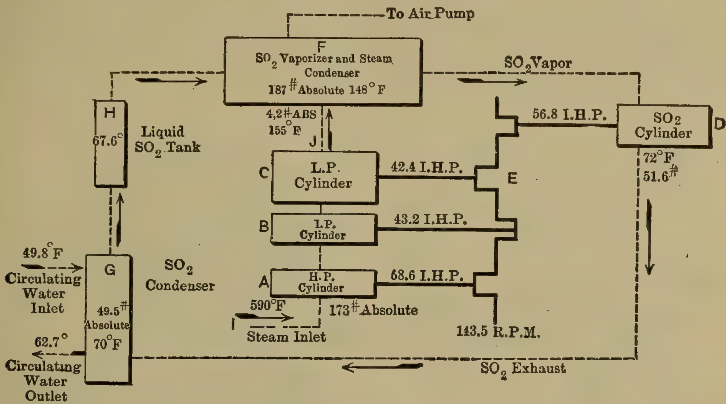


FIG. 230. Diagram of Binary-vapor Engine.

A number of experiments made by Professor E. Josse in the laboratory of the Royal Technical High School of Berlin on an experimental plant of about 200 horse power gave some remarkable results. A few of the tests made with highly superheated steam gave the following average figures:

I.h.p. (steam end)	146.4
Steam consumption per i.h.p. hour	12.8
I.h.p. (SO_2 end)	52.7
Percentage of power of SO_2 engine	35.9
Steam consumption per i.h.p. hour of combined engine	9.43

When operating under the most satisfactory conditions a performance of 8.36 pounds of steam per i.h.p. hour was recorded, corresponding to a heat consumption of 158.3 B.t.u. per minute, which is the best recorded performance to date (1912) in the history of steam-engine economy.

SO_2 does not attack the metal surface of the engine unless combined with water, in which case sulphurous acid is formed. There is, however, no danger from this cause, since the SO_2 being under greater pressure effectually prevents leakage of water into the SO_2 system.

The SO_2 cylinder requires no other lubrication than the SO_2 itself, which is of a greasy nature.

Properties of SO_2: Trans. A.S.M.E., 25-181. *Binary-Vapor Engines:* Jour. Frank. Inst., June, 1903; Elec. World and Engr., Aug. 10, 1901; U. S. Cons. Reports, No. 1139, Sept. 14, 1901; Engr. U. S., Aug. 1, 1903; Sib. Jour. of Eng., March, 1902.

212. Rotary Engines. — The rotary engine differs from the reciprocating engine in that the piston, or equivalent, rotates about the cylinder axis. Its operation is entirely different from that of the steam turbine; in the rotary engine the static pressure of the steam actuates the piston and in the turbine the momentum of the steam is imparted to the rotating element.

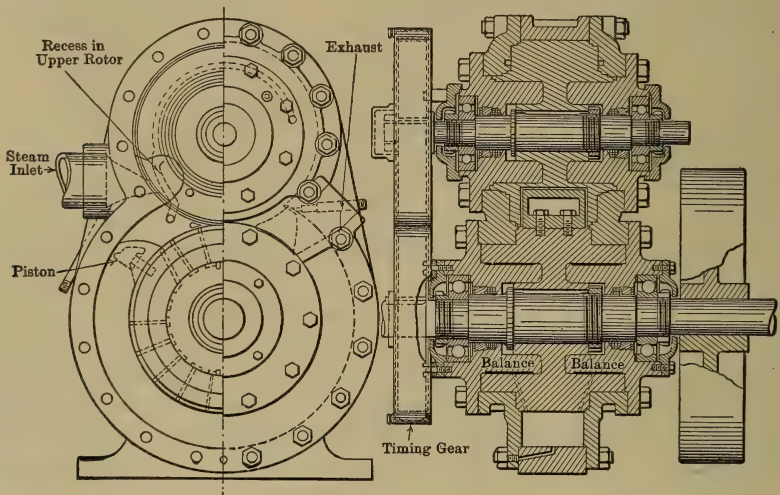


Fig. 231. Herrick Rotary Engine.

Over 2200 patents have been issued to date on rotary engines but not a single machine has yet been able to compete with the reciprocating engine as regards steam economy. The advantages of the rotary engine are many and for this reason innumerable inventors have been exerting their skill in the development of this type of prime mover, but unfortunately the impracticability of satisfactorily packing the rubbing surfaces has more than offset the advantages and the commercially successful machine is yet to be found.

The writer has tested out various types of rotary steam engines, and the best has been but a poor competitor of the ordinary grade of reciprocating mechanism.

One of the most successful rotary engines is illustrated in Fig. 231. The device consists essentially of two rotors in rolling contact, the upper one containing a recess which serves as a steam inlet and allows the piston on the lower rotor to pass, while the lower one contains the

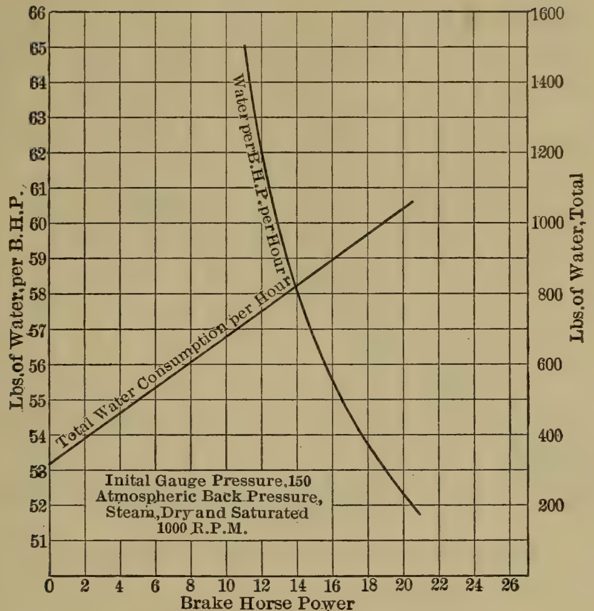


FIG. 232. Performance of Rotary Engine.

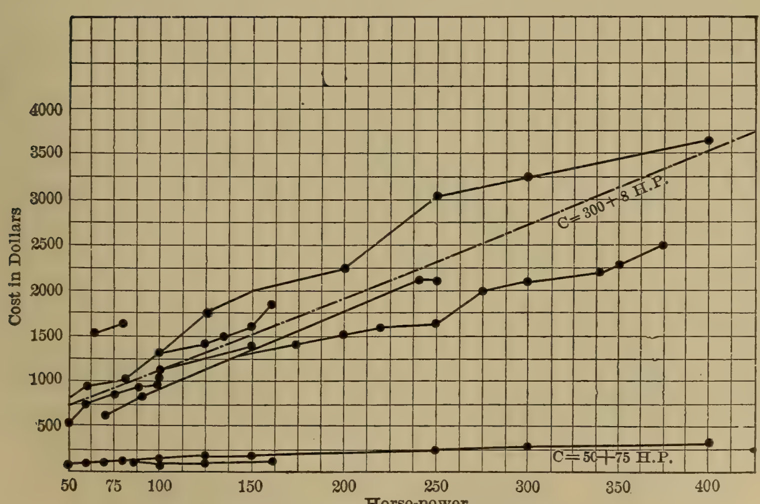


FIG. 233. Cost of Simple High-speed Engines.

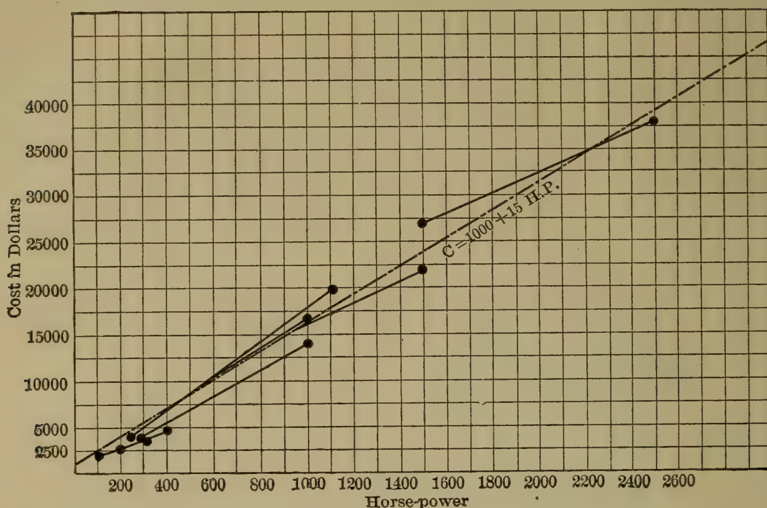


FIG. 234. Cost of High-speed Compound Engines.

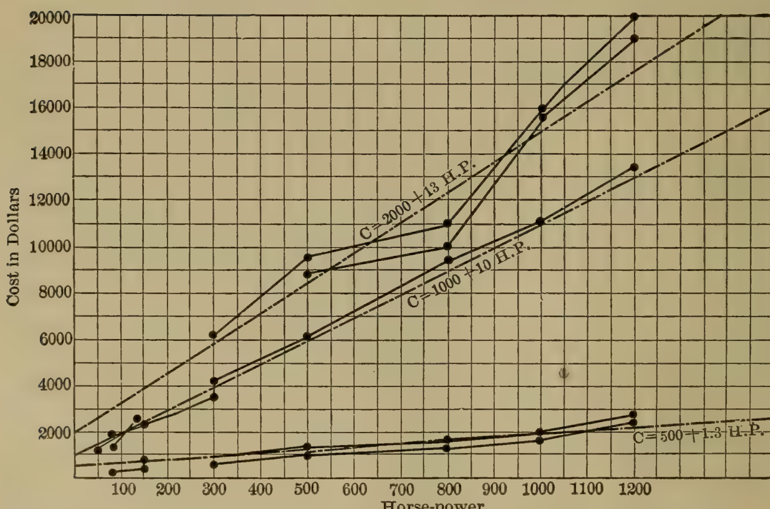


FIG. 235. Cost of Low-speed Engines, Simple and Compound.

piston and transmits the power to the shaft. In fundamental principle it is not unlike many other rotary engines in that the power is applied directly to the shaft by the expansion of steam behind a rotary piston. The synchronous movement of the two rotors is maintained by means of two timing gears on the far side of the casing. The curves in Fig. 232 are based upon the tests made by Professor Pryor of Stevens Institute of a 20-horse-power engine of this design, initial pressure 150 pounds gauge, atmospheric exhaust, steam dry and saturated.

213. Cost of Engines. — In general the cost of engines per horse power diminishes as the size increases, but is of course governed by the style and workmanship. Average figures may be expressed as follows (Engr. U. S., Nov. 15, 1902, p. 750):

Simple high-speed engines........	Cost in dollars = 300 + 8 × horse power
Setting, high-speed engines........	Cost in dollars = 60 + 0.75 × horse power
Compound high-speed engines.....	Cost in dollars = 1000 + 15 × horse power
Simple low-speed engines.........	Cost in dollars = 1000 + 10 × horse power
Compound low-speed engines......	Cost in dollars = 2000 + 13 × horse power
Setting, low-speed engines........	Cost in dollars = 500 + 1.3 × horse power

These equations were deduced from the curves in Figs. 233 to 235, which were plotted from the actual costs of a large number of engines.

Rules for testing steam engines. — See Appendix C.

For a complete description of modern types of piston engines see Prac. Engr. U. S., Jan. 1, 1913.

CHAPTER X.

STEAM TURBINES.

214. Classification. — The development of the steam turbine during the past decade has been truly remarkable. So rapid has been the growth that many turbines representative of the best practice seven years ago are virtually obsolete to-day. Because of the almost radical changes from year to year it is practically impossible to keep the descriptive features of a textbook strictly in accord with current practice, and the subject matter must necessarily be of a general nature.

Steam turbines are now being used for driving alternating-current generators, turbo-compressors, pumps, blowers and marine propellers, and, by means of gearing to furnish power for reciprocating air compressors, rolling mills and other classes of slow-speed machinery. Although the reciprocating engine will probably continue to be an important factor in the power world for years to come, its field of usefulness is being gradually limited by the steam turbine. The steam turbine has found favor chiefly on account of its low first cost, low maintenance cost, small floor-space requirements and low cost of attendance.

A general classification of steam turbines is unsatisfactory because of the overlapping of the various groups, and the following chart is offered merely as a guide in arranging a few well-known turbines according to the fundamental principles involved in their operation.

Steam Turbines	Impulse	Single Velocity.	De Laval.	Single-pressure Stage.
		Multi-velocity Stage.	Terry. Sturtevant. Curtis (Small Type). Riedler-Stumpf.	
		Single-velocity Stage.	Kerr. De Laval. Wilkinson. Rateau. Zoelly.	Multi-pressure Stage.
		Multi-velocity Stage.	Curtis.	
	Reaction.	Multi-velocity Stage.	Westinghouse-Parsons. Allis-Chalmers-Parsons.	
	Combined Impulse and Reaction.	Multi-velocity Stage.	Westinghouse Double Flow.	

As shown in the preceding chart, all turbines may be divided into three general classes, (1) impulse, (2) reaction, and (3) combined impulse and reaction, though strictly speaking all turbines depend more or less upon both impulse and reaction for their operation.

Impulse Type:

In the impulse type the steam is expanded by suitable means and the heat given up by the pressure drop imparts velocity to the jet itself. The jet impinges against the vanes of a rotating wheel and gives up its kinetic energy to the wheel. If the entire pressure drop takes place in one set of nozzles and the resulting jet is directed against a single wheel the turbine is classified with the *single-stage single-velocity group*. The velocity of the jet is very high, from 2000 to 4000 feet per second, and for satisfactory economy the peripheral velocity of the wheel must also be very high, from 700 to 1400 feet per second. The De Laval single-stage turbine is the best-known example of this group.

If the entire pressure drop takes place in a single set of nozzles and a single wheel is to be used at a comparatively low speed satisfactory economy may be effected by *compounding the velocity*. That is, the jet issuing from the nozzle at a very high velocity is reflected back and forth from the vanes on the rotor to a series of fixed reversing buckets until all of the available kinetic energy of the jet has been imparted to the wheel. The Terry single-stage turbine is representative of this group.

Low peripheral velocity and high efficiency may be obtained by *pressure compounding;* that is, expansion takes place in a series of successive nozzles instead of one nozzle. Only a part of the available heat energy is converted into kinetic energy in each set of nozzles. For each set of fixed nozzles there is a corresponding rotor. This type of turbine is to all intents and purposes a series of single-velocity impulse turbines placed side by side. The Kerr turbine is representative of this group.

By *compounding both velocity and pressure* we have the multi-velocity and pressure type of which the Curtis turbine is the best-known example.

Reaction Type:

In the reaction type the conversion of potential to kinetic energy takes place in the moving blades as well as in the fixed blades. Only a very small portion of the heat energy imparts velocity in the first set of fixed blades or nozzles. The jet issuing from this set of nozzles impinges against the first set of moving blades and imparts its kinetic energy to the rotor by *impulse*. The adjacent moving blades are proportioned so that partial expansion takes place within them and the resulting increase

in velocity exerts a *reaction* which still further accelerates the rotor. The expansion is very gradual and a large number of alternately fixed and revolving blades are necessary to effect complete expansion. Because of the small pressure drop in each stage low peripheral velocities are possible with high over-all efficiency. The Westinghouse and Allis-Chalmers designs of the Parsons turbine are the best-known examples of this type.

Combined Impulse and Reaction Type:

In this class the high-pressure elements are of the impulse type and the low-pressure elements of the reaction type. The Westinghouse-Parsons double-flow high-pressure turbine is typical of this class and is virtually a combination of the Curtis and Parsons designs. Several European impulse turbines as recently designed are fitted with reaction blades adjacent to the nozzles, showing the tendency to merge the different fundamental types.

Turbines may be classified according to the service for which they are intended, as

High-pressure non-condensing,
High-pressure condensing,
Low-pressure,
Mixed-pressure,
Bleeder.

Each of these types is discussed later on in the chapter.

Recent Developments in Steam Turbine Practice: Mech. Engr., Jan. 26, 1912.
The Present State of Development of Large Steam Turbines: Jour. A.S.M.E., May, 1912.
The Steam Turbine: Engng., Dec. 29, 1911.
Status of the Small Steam Turbine: Power, Jan. 2, 1912.

215. General Elementary Theory. — A given weight of steam at a given pressure and temperature occupies a certain known volume and contains a known amount of heat energy. If the steam is permitted to expand to a lower pressure without receiving additional heat or giving up heat to surrounding bodies it is capable of doing a certain amount of work which will be the same whether the expansion takes place in the cylinder of a reciprocating piston engine, a rotary piston engine, or the nozzles and blades of a steam turbine.

Let W = weight of steam, lbs. per sec.
E = energy given up by 1 lb. of steam, ft.-lbs.
P_1 = initial pressure, lbs. per sq. in. abs.
P_n = final pressure, lbs. per sq. in. abs.
H_1 = initial heat content per lb., B.t.u.
H_n = final heat content per lb., B.t.u.

Then the heat drop, or heat available for doing useful work, is

$$W\,(H_1 - H_n)\text{ B.t.u.} \tag{124}$$

If the steam expands against a resistance, as, for example, the piston of a reciprocating engine, the energy given up in forcing the piston forward may be expressed

$$E_1 = 777.5\,W\,(H_1 - H_n)\text{ ft.-lbs.} \tag{125}$$

If the steam expands within a perfect nozzle the energy will be given up in imparting velocity to the steam itself, thus:

$$E_2 = W\frac{V_1^2}{2\,g}\text{ ft.-lbs.} \tag{126}$$

in which

V_1 = velocity of the jet in feet per second.

If the velocity of the jet is retarded to V_n feet per second, as by placing a series of vanes in its path, then the energy given up to the vanes (neglecting all losses) is

$$E = W\frac{V_1^2 - V_n^2}{2\,g}. \tag{127}$$

If the jet is brought to rest by the vanes (neglecting all losses), then $V_n = 0$ and the energy given up is

$$E_3 = W\frac{V_1^2}{2\,g}. \tag{128}$$

But $E_1 = E_3$. Hence,

$$777.5\,W\,(H_1 - H_n) = W\frac{V_1^2}{2\,g},$$

from which

$$V_1 = 223.8\sqrt{H_1 - H_n}.^* \tag{129}$$

If there are n pressure stages, then the theoretical stage velocity is

$$V_1' = 223.8\sqrt{\frac{H_1 - H_n}{n}}. \tag{130}$$

The jet issuing from the nozzle is capable of exerting an *impulse* equal to F upon any object in its path, thus:

$$F = \frac{WV_1}{g}\text{ lbs.} \tag{131}$$

If A = the area of cross section of the jet in square feet, and γ = weight of steam, pounds per cubic foot, then $W = \gamma A V_1$, or

$$F = \frac{\gamma A V_1^2}{g}\text{ lbs.} \tag{132}$$

* For most purposes it is sufficiently accurate to make 223.8 = 224.

The *reaction*, R, of the jet against the nozzles is equal in value and opposite in direction to the impulse, or

$$R = F = \frac{WV_1}{g} = \frac{\gamma A V_1^2}{g}. \tag{133}$$

The theoretical horse power developed by a jet of steam flowing at the rate of one pound per second may be expressed

$$\text{H.P.} = \frac{E}{550} = \frac{V_1^2 - V_n^2}{2g \times 550}, \tag{134}$$

in which

V_1 = initial velocity of the jet, ft. per sec.
V_n = final velocity of the jet, ft. per sec.

Steam consumption per horse-power hour:

$$W_1 = \frac{3600}{\text{H.P.}}. \tag{135}$$

Heat consumption, B.t.u. per horse power, per minute:

$$= \frac{W(H_1 - q_n)}{60}, \tag{136}$$

in which

q_n = heat of the liquid at pressure P_n.

Impulse efficiency of the jet = equation (127) ÷ equation (128).

$$E_i = \frac{V_1^2 - V_n^2}{V_1^2}. \tag{137}$$

Thermal efficiency (Rankine Cycle):

$$E_r = \frac{H_1 - H_n}{H_1 - q_n}. \tag{138}$$

Efficiency ratio or "kinetic" efficiency:

$$E = \frac{2546}{W_1(H_1 - H_n)}. \tag{139}$$

Equations (124) to (139) are general and are applicable to all turbines of whatever make.

The more important types of turbines will be discussed separately and an application of above equations will be made in each specific case.

Heat Drop in Steam Turbines: Trans. A.S.M.E., Vol. 33, p. 325, 1911; Engr., Mar. 8, 1912.

216. The De Laval Turbine. — Fig. 236 shows a section through a De Laval steam turbine and gear case and illustrates the principles of the single-stage "impulse" type. The turbine proper, to the right of the figure, consists of a high-carbon steel disk C fitted at the periphery with a single row of drop-forged steel blades and inclosed in a cast-steel

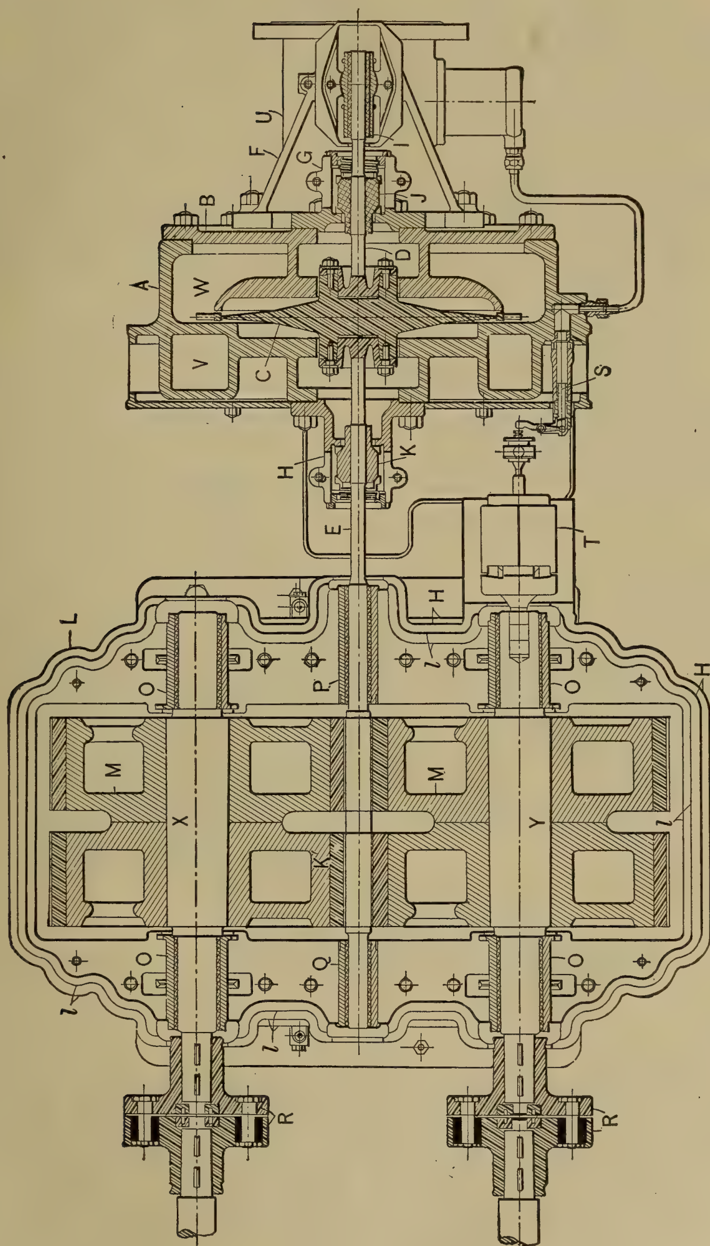


FIG. 236. De Laval Turbine, Double Gear, Single Stage.

casing. The disk is secured to a light flexible shaft and is of such a cross section that the radial and tangential stresses throughout its mass are of constant value. A flexible shaft is employed which allows the wheel to assume its proper center of rotation and thus to operate like a truly balanced rotating body.* The shaft is supported by three bearings, *P*, *K*, and *I*. *I* is self-aligning and carries the greater part of the weight of the disk. *K* is a flexible bearing, entirely free to oscillate with the shaft, and its only function is to seal the wheel casing against leakage. The power is transmitted through a steel helical pinion *K'* mounted on the extension of the turbine shaft *E*, to two large gears *M*, *M* at a reduction in speed of about 10 to 1. The blades, Fig. 237, are made with a bulb shank and fitted in slots milled in the rim of the wheel. The flanges, at the outer end of the blades, are brought in contact with each other and calked so as to form a continuous ring. The inlet and outlet angles of the blades are made alike and are 32 degrees for smaller sizes and 36 degrees for larger sizes.

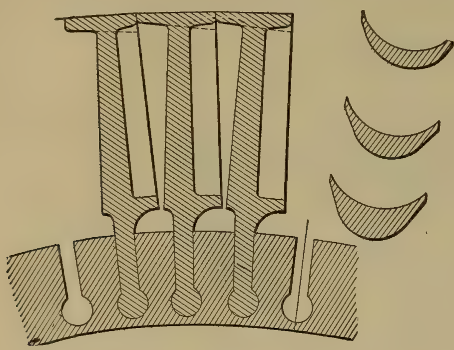


Fig. 237. De Laval Blades.

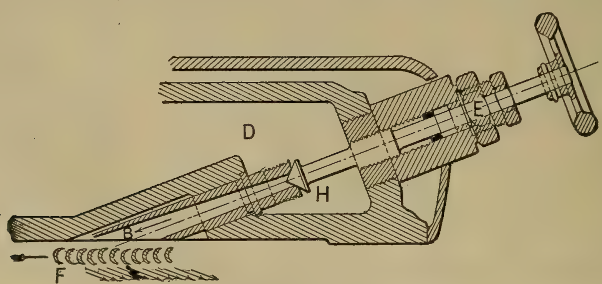


Fig. 238. De Laval Nozzle.

The operation is as follows: Steam enters the steam chest *D*, Figs. 236 and 237, through the governor (shown in detail in Fig. 239) and is distributed to the various adjustable nozzles, varying in number from 1 to 15 according to the size of turbine. In the earlier types the nozzles were uniformly distributed around the circumference, but in the later types are arranged in groups. As illustrated in Fig. 238

* The shaft diameter for a 100-horse-power turbine is but 1 inch and for a 300-horse-power turbine approximately $1\frac{15}{16}$ inches.

the nozzles are placed at an angle of 20 degrees with the plane of the disk. The steam is expanded adiabatically in the nozzles to the existing back pressure before it impinges at high velocity against the blades. After giving up its energy the steam passes into chamber *W*, Fig. 236, and out through the exhaust opening. Fig. 239 gives the details of the governor and vacuum valves. Two weights *B* are pivoted on knife edges *A* with hardened pins *C* bearing on the spring *D*. *E* is the governor body, fitted in the end of the gear-wheel shaft *K*, and has seats milled for the knife edges *A*. The spring seat *D* is held against pins *A* by

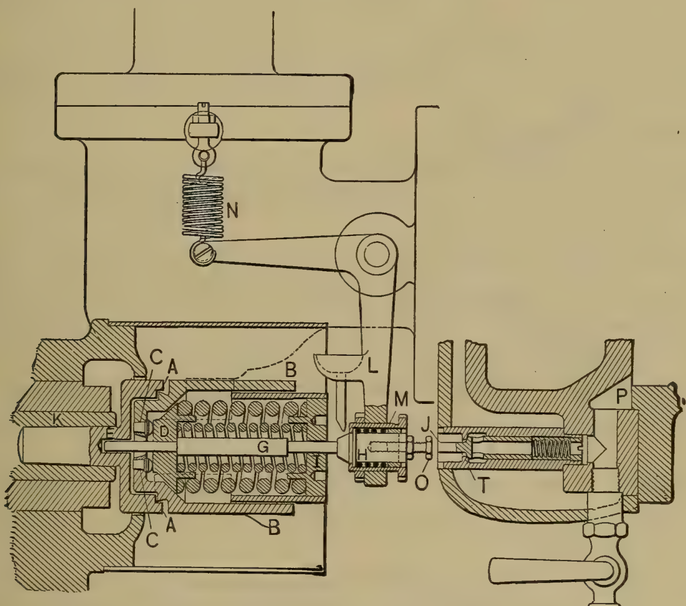


FIG. 239. De Laval Governor.

spiral concentric springs, the tension on which is adjusted by a milled nut *I*. When the speed exceeds the normal, centrifugal force causes the weights to fly outward and overcome the resistance of the springs. This pushes pin *G* against bell crank *L*, which in turn closes the double-seated valve, thus throttling the supply of steam. To prevent racing in case the load is suddenly removed the vacuum valve *T* is added to the governor mechanism. Its operation is as follows: The governor pin *G* actuates the plunger *H* under normal conditions without moving the plunger relative to the bell crank. In case the load is suddenly removed, centrifugal force pushes pin *G* against bell crank *L* until it reaches its extreme position and the valve is nearly closed and little steam enters the turbine. If this does not check the speed, plunger *G*

overcomes the resistance of spring M, and H moves relative to L, and its adjustable projection O presses against valve stem T and allows air to rush into the turbine chamber through passage P.

The power of the turbine depends upon the number of nozzles in action, and these can be opened or closed by a hand wheel on each. Each nozzle performs its function as perfectly when operating alone as when operating in conjunction with others.

De Laval turbines of the single-stage geared type are made in sizes ranging from 17 to 700 horse power, condensing and non-condensing, and are designed to regulate within an extreme variation of 2 per cent from no load to full load.

The speeds vary from 10,600 r.p.m. for the largest size to 30,000 r.p.m. for the smallest, the gearing reducing these to 900 and 3000 r.p.m., respectively, at the shaft.

The diameter of the wheel varies from 4 inches in the smallest turbine to 30 inches in the largest, thus giving peripheral velocities of from 520 to 1310 feet per second.

In addition to the single-stage geared type the De Laval company manufacture the single-stage gearless, two-stage gearless, and multi-stage gearless turbine. The latter are constructed in sizes up to 5000 kilowatts and are described in paragraph 221.

217. Elementary Theory. — De Laval Single-stage Turbine. — The maximum theoretical power developed by a jet of steam flowing through a nozzle is dependent only upon the *weight* of steam flowing per unit of time and the *initial velocity*. Therefore the higher the initial velocity for a given rate of flow the greater will be the power developed and the higher the efficiency.

The maximum *weight* of steam discharged through a nozzle of any shape and for a given initial pressure is determined by the *area* of the narrowest cross section or *throat*.

To obtain the maximum *velocity* at the exit or *mouth*, for a given rate of flow, the nozzle should be proportioned so that expansion to the external pressure into which the nozzle delivers shall take place within the nozzle itself. If expansion in the nozzle is incomplete, sound waves will be produced and there will be irregular action and loss of energy. On the other hand, if expansion in the nozzle is carried below that of the external pressure at the mouth, sound waves will be produced with subsequent loss of energy even greater than in the former case.

Experimental and mathematical investigations indicate that the pressure at the narrowest section of an orifice or the throat of a nozzle through which steam is flowing falls to approximately 0.58 of the initial absolute pressure (with resultant velocity of about 1400 to 1500 feet per

second) and any farther fall in pressure must take place beyond the narrowest section. Thus for back pressures greater than 0.58 of the initial (conveniently taken as $\frac{3}{5}$), maximum exit velocity may be obtained from orifices or nozzles of uniform cross section or with sides *convergent.* For back pressure less than 0.58 of the initial the nozzle must first *converge* from inlet to throat and then *diverge* from throat to mouth in order to obtain maximum velocity. Without the divergent portion of the nozzle the jet will begin to spread after passing the throat, and its energy will be given up in directions other than that of the original jet.

Fig. 240 shows a section through a theoretically proportioned expanding nozzle. The cross section of the tube at any point n may be

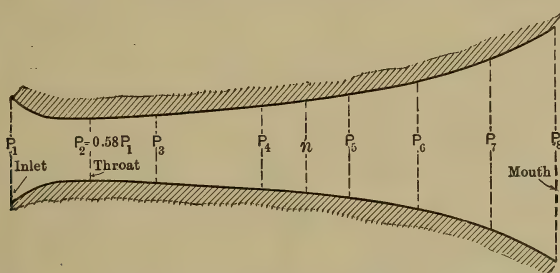


FIG. 240. Theoretically Proportioned Expanding Nozzle.

calculated by means of equation

$$A_n = \frac{WS_n}{V_n}, \tag{140}$$

in which

A_n = area in square feet.

W = maximum weight of steam discharged, pounds per second.

S_n = specific volume of the steam at pressure P_n.

For saturated steam $S_n = x_n u_n$,

in which x_n = quality of steam at pressure P_n after adiabatic expansion from pressure P_1.

u_n = specific volume of saturated steam at pressure P_n.

For superheated steam, see Mollier diagram, Appendix L.

V_n = velocity of the jet, feet per second.

V_n may be determined from equation (129):

$$V_n = 223.8 \sqrt{H_1 - H_n}.$$

By substituting H_n = heat content corresponding to pressure $P_n = 0.58\, P_1$ in equations (129) and (140) the area at the throat may be readily determined. The cross-sectional area for other points in the

tube may be determined in a similar manner by assigning values of H_n corresponding to the various pressures.

In case of a perfect nozzle $H_1 - H_n$ represents the heat given up toward producing velocity by adiabatic expansion from pressure P_1 to P_n. In the actual nozzle the frictional resistance of the tube serves to increase its dryness fraction, but in doing so it decreases the amount of energy the steam is capable of giving up towards increasing its own

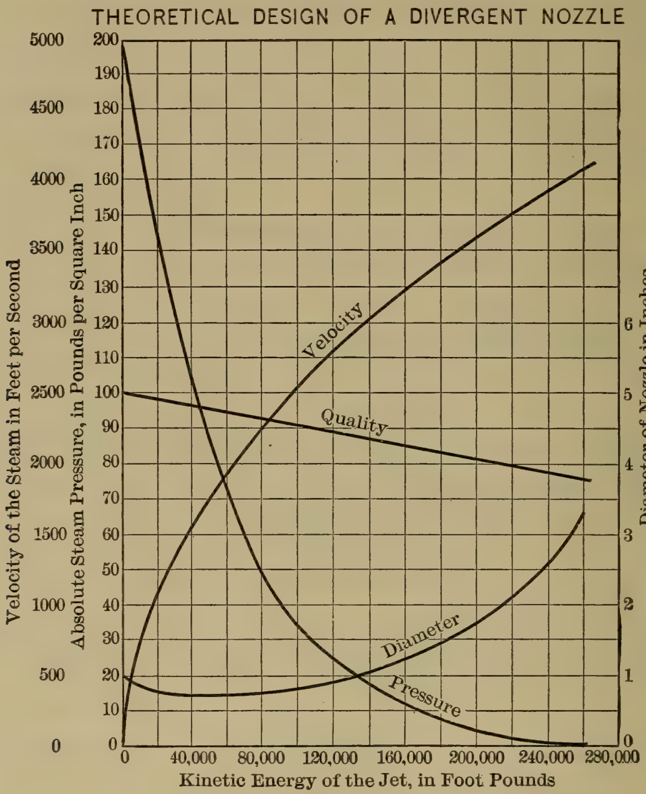


FIG. 241.

velocity. If y one-hundredths of the heat $H_1 - H_n$ is utilized in overcoming frictional resistance, then the resulting velocity will be

$$V = 223.8 \sqrt{(1 - y)(H_1 - H_n)}. \tag{141}$$

The quality of the steam after expanding to P_n against the resistance will be higher by an amount

$$I_n = \text{increase in quality} = \frac{y(H_1 - H_n)}{r_n}, \tag{142}$$

in which

$$r_n = \text{heat of vaporization at pressure } P_n.$$

The curves in Fig. 241, calculated by means of equations (129) and (140), show the relationship between velocity, quality, pressure, and kinetic energy for all points in a theoretically perfect nozzle expanding one pound of dry steam per second from an initial absolute pressure of 190 pounds to a condenser pressure of one pound.

The curves in Fig. 242 are based upon the experiments of Gutermuth (Zeit. d. Ver. Ingr., Jan. 16, 1904) and show the effect of a few shapes of nozzles and orifices on the actual weight of steam discharged for various rates of initial and final pressures, the smallest section of the tube remaining constant.

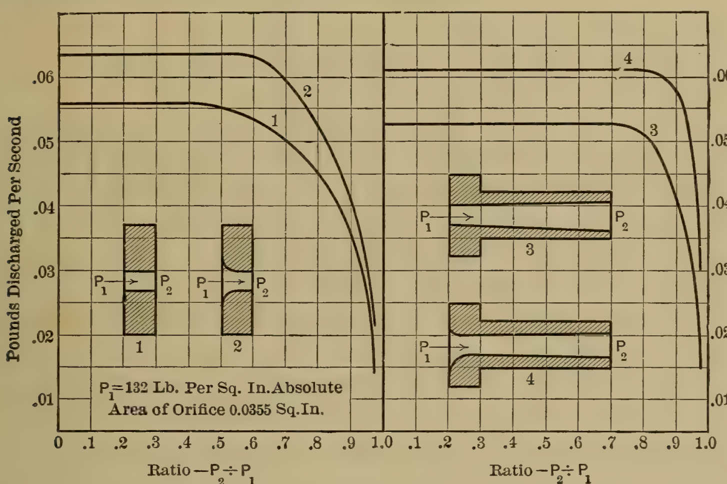


FIG. 242. Flow of Steam through Nozzles.

The nozzles of most commercial types of steam turbines are made with straight sides as in Fig. 238, so that only the area at the mouth need be determined in addition to that at the throat in order to lay out the shape of the tube.

Equations (129) and (140) are general and are applicable to steam of any quality, wet, dry, or superheated. For steam initially dry and saturated *Napier's* rule offers a simple means of determining the area at the throat, thus:

$$W = \frac{A_0 P_1}{70} \text{ for } P_n = \text{ or } < \frac{3}{5} P_1, \tag{143}$$

$$W = 0.029\, A_0 \sqrt{P_n (P_1 - P_n)} \text{ for } P_n > \tfrac{3}{5} P_1, \tag{144}$$

in which

W = maximum weight of steam discharged, pounds per second;
A_0 = area at the throat, square inches;
P_1 = absolute initial pressure, pounds per square inch;
P_n = absolute back pressure, pounds per square inch.

Moyer ("The Steam Turbine," 1st Edition, p. 40) states that the ratio of the area of a correctly proportioned nozzle at the throat A_0 to the area at any point A_n is very nearly proportional to the ratio of the pressure at point A_n to the initial pressure, or

$$\frac{A_0}{A_n} = \frac{P_1}{P_n}. \tag{145}$$

The entrance to the tube is rounded by any convenient curve.

The length of the tube may be roughly approximated by the following formula:

$$L = \sqrt{15\,A_0}, \tag{146}$$

in which

L = length between the throat and mouth, in inches;
A_0 = area at the throat, square inches.

Practice shows that the cross section of a nozzle, whether circular, elliptical, square, or rectangular (the latter with rounded corners), has very little influence on the efficiency provided the inner surfaces are smooth and the ratio of the area at the throat to that of the mouth is correctly proportioned. The *velocity* efficiency of a properly proportioned nozzle with straight sides is about 95 to 97 per cent, corresponding to an *energy* efficiency of 92 to 94 per cent, so that it is not considered worth while to attempt to follow the more difficult exact curves.

Example: — Find the smallest cross section of a frictionless conically divergent nozzle for expanding one pound of steam per second from an absolute initial pressure of 190 pounds to an absolute back pressure of 2 pounds and find six intermediate cross sections where the pressures will be 70, 30, 14.7, 8, 4, and 2 lbs. respectively. Compare the velocity and energy of the jet issuing from this nozzle with those of an actual nozzle in which 10 per cent of the heat energy is lost in friction.

From steam and entropy tables we find the values of H, x, u, for absolute pressures corresponding to 190, 0.58 × 190 = 110, 70, 30, etc., lbs. per square inch as follows (theoretical nozzle):

	H.	x.	u.	$S = xu$.
P_1 = 190	1197.3	1.00	2.405	2.394
P_2 = 110*	1152.6	0.960	4.047	3.878
P_3 = 70	1117.9	0.932	6.199	5.775
P_4 = 30	1057.2	0.887	13.75	12.27
P_5 = 14.7	1011.3	0.857	26.78	22.95
P_6 = 8	947.8	0.834	47.26	39.29
P_7 = 4	935.6	0.810	90.4	73.2
P_8 = 2	899.3	0.788	173.1	137.0

* $P_2 = 0.58\,P_1$ (= pressure at throat).

If entropy tables or charts are not available, values H_1 to H_8 and x_1 to x_8 must be calculated.

The different quantities for the theoretical nozzle will be calculated for the exit pressure $P_n = P_8 = 2$ lbs. per sq. in. absolute.

$$\begin{aligned} V_8 &= 223.8\sqrt{H_1 - H_8} \\ &= 223.8\sqrt{1197.3 - 899.3} \\ &= 3865 \text{ feet per second.} \\ E_8 &= 778\,(H_1 - H_8) \\ &= 778\,(1197.3 - 899.3) \\ &= 232{,}000 \text{ foot-pounds.} \\ A_8 &= \frac{WS}{V} \\ &= \frac{1 \times 137}{3865} \\ &= 0.0353 \text{ square foot.} \\ d_8 &= \sqrt{\left(\frac{144 \times 4}{\pi}\right)A} = 13.56\sqrt{A} \\ &= 13.56\sqrt{0.0353} \\ &= 2.54 \text{ inches.} \\ F_8 &= \frac{WV_8}{g} \\ &= \frac{3865}{32.2} \\ &= 120 \text{ pounds.} \end{aligned}$$

THEORETICAL NOZZLE.

Quantity		V Ft. per Sec.	E Ft.-Lbs.	A Sq. Ft.	d Inches.	F Pounds.
Formula		(73)	(72)	(76c)		(74)
Pressures	110	1,496	34,767	.00259	0.693	46.4
	70	1,995	61,853	.00269	0.702	61.98
	30	2,650	107,485	.00461	0.919	82.3
	14.7	3,053	144,742	.00745	1.1	94.8
	8	3,339	173,207	.0119	1.46	103.7
	4	3,624	203,968	.0202	1.92	112.5
	2	3,865	232,000	.0353	2.54	120.0

In the actual nozzle these values will be modified because of the frictional losses. Thus, for $P_n = 2$ lbs.,

$$\begin{aligned} V_8 &= 223.8\sqrt{(1 - y)\,(H_1 - H_8)} \\ &= 223.8\sqrt{(1 - 0.1)\,(1197.3 - 899.3)} \\ &= 3667 \text{ ft. per sec.} \end{aligned}$$

$$E_8 = 778\,(1 - 0.1)(1197.3 - 899.3) = 208{,}800 \text{ ft.-lbs.}$$

$$x_8' = x_8 + I_8 = x_8 + \frac{y\,(H_1 - H_8)}{r_8}$$

$$= 0.788 + \frac{0.1\,(1197.3 - 899.3)}{1021}$$

$$= 0.788 + 0.029$$

$$= 0.817.$$

$$A_8 = \frac{W x_8' u_8}{V_8}$$

$$= \frac{0.817 \times 173.1}{3667}$$

$$= 0.0386 \text{ sq. ft.},$$

from which

$$d_8 = 2.66 \text{ in.}$$

$$F_8 = \frac{W V_8}{g} = \frac{3668}{32.2} = 114 \text{ lbs.}$$

These various factors for all given pressures have been calculated in a similar manner and are as follows:

ACTUAL NOZZLE.

Quantities		V Ft. per Sec.	E Ft.-Lbs.	x'.	A Sq. Ft.	d Inches.	F Ft.-Lbs.
Pressures	110	1,420	31,317	.9658	.00275	0.711	44.1
	70	1,893	55,632	.9414	.00286	0.723	58.8
	30	2,515	98,257	.9026	.00493	0.951	78.12
	14.7	2,894	130,050	.876	.0080	1.2	98.8
	8	3,168	155,858	.856	.0127	1.53	98.4
	4	3,438	183,581	.836	.0220	2.01	106.8
	2	3,667	208,800	.817	.0386	2.66	114.0

Many of these values may be determined directly from the *Mollier* or total heat-entropy diagram as described in Appendix L; in fact, the Mollier diagram has to all intents and purposes supplanted the steam tables in this connection. For superheated steam the diagram is extremely useful in avoiding laborious calculations.

Fig. 243 gives a diagrammatic arrangement of the blades in a single-stage De Laval turbine. The nozzle directs the steam against the blades with *absolute* velocity V_1 and at an angle α with the plane of the wheel XX. Since the wheel is moving at a velocity of u feet per second, the velocity v_1 of the steam *relative* to the wheel is the resultant of V_1 and u. The angle β_1 between v_1 and XX will be the proper blade angle at entrance. If the blade curve makes this angle with the direction of motion of the wheel no shock will be experienced when the steam enters

the blades. For convenience in construction the exit angle β_2 is made the same as the entrance angle β_1. Neglecting frictional losses in the blade channels the *relative* exit velocity will be $v_2 = v_1$, and the *absolute* velocity V_2 is the resultant of v_2 and u. The impulse exerted by the jet in striking the vanes is $\frac{W}{g}v_1$, and its component in the direction of motion is $\frac{W}{g}v_1 \cos\beta_1 = \frac{W}{g}(V_1 \cos\alpha - u)$. As the jet leaves the vanes the impulse is $-\frac{W}{g}v_2 \cos\beta_2 = -\frac{W}{g}(V_2 \cos\gamma + u)$.

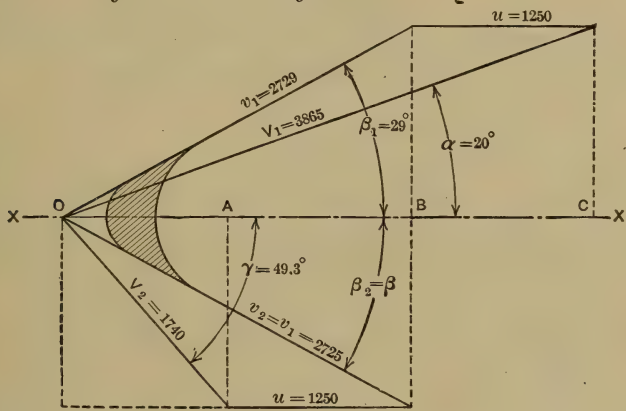


FIG. 243. Velocity Diagram. Ideal Single-stage Impulse Turbine.

The *total pressure* acting on the vanes, or the actual driving impulse, is

$$P = \frac{W}{g}\left\{V_1 \cos\alpha - u - \left[-(V_2 \cos\gamma + u)\right]\right\}$$

$$= \frac{W}{g}(V_1 \cos\alpha + V_2 \cos\gamma). \qquad (147)$$

Equation (147) may also be expressed

$$P = \frac{W}{g}\cdot 2\,(V_1 \cos\alpha - u). \qquad (148)$$

The resultant *axial force* or *end thrust* is

$$F = \frac{W}{g}(V_1 \sin\alpha - V_2 \sin\gamma). \qquad (149)$$

Evidently if $\alpha = \gamma$ and $V_1 = V_2$ there will be no end thrust, since $V_1 \sin\alpha - V_2 \sin\gamma$ will be zero.

The *work done* is

$$Pu = \frac{W}{g}u\,(V_1 \cos\alpha + V_2 \cos\gamma), \qquad (150)$$

or, using equation (148) in place of (147),

$$Pu = \frac{W}{g} \cdot 2u\,(V_1 \cos\alpha - u)$$
$$= \frac{W}{g} \cdot 2\,(uV_1 \cos\alpha - u^2). \tag{151}$$

By making the first derivative equal to zero

$$\frac{d}{du}\left\{\frac{W}{g} 2\,(uV_1 \cos\alpha - u^2)\right\} = V_1 \cos\alpha - 2u = 0,$$

or
$$u = \tfrac{1}{2} V_1 \cos\alpha.$$

That is, for *any nozzle angle* α the work done, Pu, has its greatest value when $u = \frac{1}{2} V_1 \cos\alpha$ or $\gamma = 90$ degrees, whence

$$Pu = W \frac{V_1^2}{2g} \cos^2\alpha. \tag{152}$$

The work for *any initial velocity* V_1 becomes a maximum when $\alpha = 0$ and $u = \frac{1}{2} V_1$. *This condition can only occur for a complete reversal of jet and zero final velocity.* Substitute $\alpha = 0$ and $u = \frac{1}{2} V_1$ in equation (151).

$$Pu = \frac{WV_1^2}{2g},$$ which is necessarily the same as equation (128).

In the actual turbine the various velocities will be less than those as obtained on account of the frictional resistance in the blades, and the velocity diagram should be modified accordingly.

Example: Lay out the blades (theoretical and actual) for the nozzle in the preceding example, assuming that the jet impinges against the wheel at an angle of 20 degrees and that the peripheral velocity is 1250 feet per second.

Theoretical Case:

Lay off $V_1 = 3865$ feet per second in direction and amount as shown in Fig. 243 and combine it with $u = 1250$ feet per second; this gives v_1, the relative entrance velocity, as 2725 feet per second, and β, the entrance angle, as 29 degrees.

Lay off $v_2 = v_1$ at an angle $\beta_2 = \beta_1$ and combine with u; this gives V_2, the *absolute* exit velocity, as 1740 feet per second.

The theoretical energy available for doing work is

$$E = \frac{W}{2g}(V_1^2 - V_2^2)$$
$$= \frac{1}{64.4}(3865^2 - 1740^2) \doteq 185{,}000 \text{ foot-pounds.}$$

The difference between 232,000 and 185,000 = 47,000 foot-pounds is evidently the kinetic energy lost in the exhaust due to the exit velocity.

The pressure exerted by the steam on the buckets is

$$P = \frac{W}{g}(V_1 \cos \alpha + V_2 \cos \gamma)$$
$$= \frac{1}{32.2}(3865 \times 0.9397 + 1740 \times 0.65166)$$
$$= 148 \text{ pounds.}$$

The theoretical impulse efficiency is

$$\frac{V_1^2 - V_2^2}{V_1^2} = \frac{3865^2 - 1740^2}{3865^2} = 0.797.$$

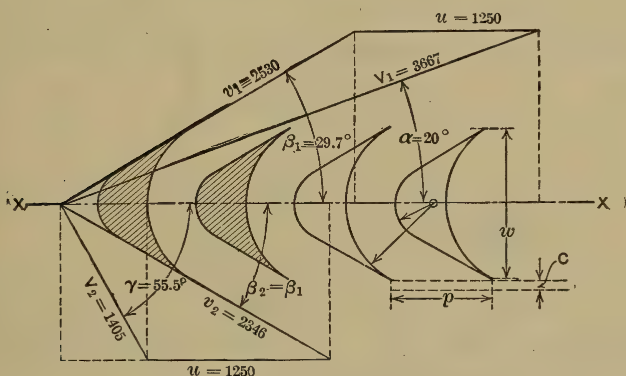


FIG. 244. Velocity Diagram as Modified by Friction Losses.

The theoretical horse power per pound of steam flowing per second is

$$\text{H.P.} = \frac{185{,}000}{550} = 336.$$

Theoretical steam consumption per horse-power hour is

$$\frac{3600}{336} = 10.7 \text{ pounds.}$$

Actual Case:

Proceed as in the theoretical case, using the actual absolute velocity $V_1 = 3865\sqrt{1-y} = 3865\sqrt{1-0.10} = 3667$ feet per second in place of the theoretical value $V_1 = 3865$. Lay off $V_1 = 3667$ at an angle of 20 degrees as before and combine with $u = 1250$, Fig. 244.

The resultant $v_1 = 2530$ is the velocity of the jet relative to the wheel, and the entrance angle β is found to be 29.7 degrees. The relative exit velocity v_2 will be less than v_1 because of the blade friction.

Assume the loss of energy ϕ from this cause to be 14 per cent; then, since the velocity varies as the square root of the energy,

$$v_2 = v_1 \sqrt{1 - \phi} \tag{153}$$
$$= 2530 \sqrt{1 - 0.14}$$
$$= 2346 \text{ feet per second.}$$

The resulting absolute velocity V_2 is found from the diagram to be $V_2 = 1405$ feet per second.

Since the loss of energy in the nozzle is

$$\frac{V_1^2 - (1 - y) V_1^2}{2g}, \tag{154}$$

and that in the blade

$$\frac{v_1^2 - (1 - \phi) v_1^2}{2g}, \tag{155}$$

the remaining energy, deducting both losses in the nozzle and the blades, is

$$\frac{W}{2g}(V_1^2 - yV_1^2 - \phi v_1^2 - V_2^2) \tag{156}$$
$$= \frac{1}{64.4}(3865^2 - 0.1 \times 3865^2 - 0.14 \times 2530^2 - 1405^2)$$
$$= 164{,}200.$$

The losses due to windage, leakage past the buckets and mechanical friction must be deducted from these figures to give the actual energy available for doing useful work. Assuming a loss of 15 per cent due to this cause, the work delivered is

$$0.85 \times 164{,}200 = 139{,}570 \text{ foot-pounds.}$$

The efficiency in the ideal case was found to be 0.797 and the available energy 185,000 foot-pounds.

The efficiency, deducting the loss due to friction, etc., is

$$\frac{139{,}570}{185{,}000} \times 0.797 = 0.60.$$

The horse power delivered is

$$\frac{139{,}570}{550} = 254.$$

Steam consumption per horse-power hour is

$$\frac{3600}{254} = 14.2 \text{ pounds.}$$

The heat consumption, B.t.u. per horse power, per minute is

$$\frac{14.2\,(1197.3 - 94)}{60} = 260.$$

Assuming the revolutions per minute to be 10,000, the mean diameter of the wheel to give a peripheral velocity of 1250 feet per second is

$$\frac{1250 \times 60}{10{,}000 \times 3.14} = 2.39 \text{ feet, or } 28.6 \text{ inches.}$$

The determination of the height and width of vanes, clearance between nozzles and blades, etc., are beyond the scope of this work and the reader is referred to the accompanying bibliography.

Blade Design for De Laval Turbines: Moyer, "Steam Turbine," Chap. IV; Power, Mar. 17, 1908, p. 391.

Flow of Steam through Nozzles: Jour. A.S.M.E., Mid. Nov., 1909, April, 1910, p. 537; Engineering, Feb. 2, 1906; Engr., Lond., Dec. 22, 1905; Eng. Rec., Oct. 26, 1901; Power, May, 1905; Eng. News, Sept. 19, 1905, p. 204.

Design of Turbine Disks: Engr., Lond., Jan. 8, 1904, p. 34, May 13, 1904, p. 481.

Turbine Losses and their Study: Jour. El. Power and Gas, March 9, 1912.

Critical Velocity of Shafting: Jour. A.S.M.E., June, 1910, p. 1060; Power, Sept., 1903, p. 484.

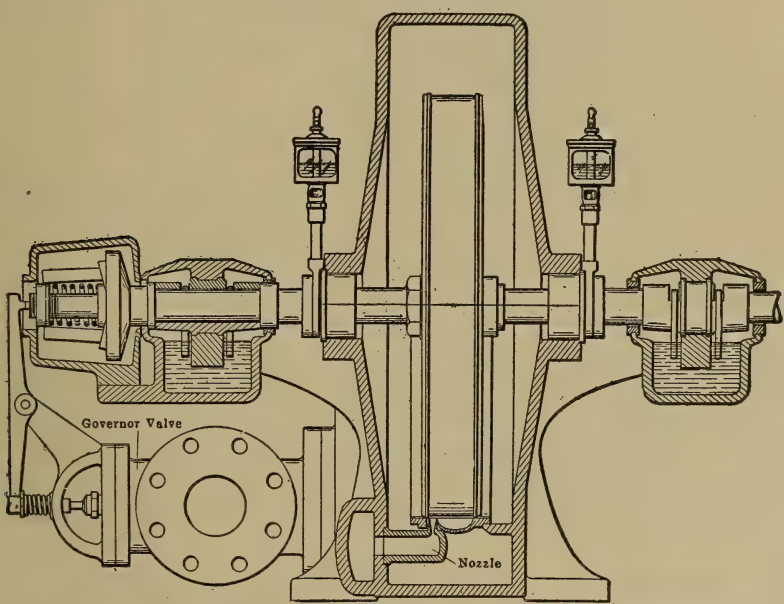


Fig. 245. Section through Single-stage Terry Steam Turbine.

218. Terry Turbine. — Fig. 245 shows a section through a single-stage Terry turbine, illustrating an application of the *single-stage impulse* type with two or more *velocity stages.* This "compounding" of the velocity permits of much lower peripheral velocities than with the single-velocity type. The rotor, a single wheel consisting of two steel disks held together by bolts over a steel center, is fitted at its periphery

with pressed-steel buckets of semi-circular cross section. The inner surface of the casing is fitted with a series of gun-metal reversing buckets arranged in groups, each group being supplied with a separate nozzle. The steam issuing from nozzle N, at very high velocity, Fig. 246, strikes one of the buckets, B, on the wheel, and since the velocity of the buckets is comparatively low, is reversed in direction and directed into the first one of the reversing chambers. The chamber redirects the jet against the wheel, from which it is again deflected; this is repeated four or more times until the available energy has been absorbed by the rotor. Terry turbines are made in a number of sizes varying from 5 to 800 horse power, and operate at speeds varying from 210 feet per second in the smaller machine to 260 feet per second in the larger.

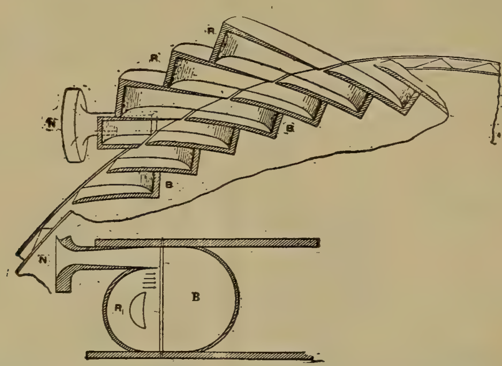


Fig. 246. Arrangement of Buckets and Reversing Chambers in a Terry Steam Turbine.

These low speed limits compared with the speed of single-stage De Laval turbines are made possible by the application of the velocity-stage principle in the use of the reversing buckets. The rotor of the smaller machine is 12 inches in diameter and runs at 3800 r.p.m., and that of the larger, 48 inches, running at 1250 r.p.m. Since the flow of steam into and from the buckets is in the plane of the wheel there is no end thrust.

Non-condensing Terry turbines are all of the single-stage type. The condensing units are of the two-stage type, the first stage expanding the steam from initial to atmospheric pressure and the second stage expanding the steam from atmospheric to condenser pressure.

For a description of the Bliss, Dake, Sturtevant and Wilkinson steam turbines with results of tests see "Small Steam Turbines," by G. A. Orrok, Jour. A.S.M.E., May, 1909, and contributed discussion, Sept., 1909. See also "The Development of the Small Steam Turbine," Eng. Mag., Dec., 1908, and Jan., 1909.

219. Elementary Theory. — Terry Turbine. — Fig. 247 gives the theoretical velocity diagram for a single pressure stage Terry Turbine. Since the entire heat drop takes place in the nozzle the initial velocity of the jet OA is the same as with the single-stage De Laval turbine and may be calculated by means of equation (129). OA represents the absolute velocity of the jet, OC the peripheral velocity and AOC the angle of the nozzle. CB is the component, parallel to the line of the jet, of the resultant of AO and OC. DC, in line with and equal in length to CB, combined with the peripheral velocity DE gives EC, the absolute velocity of the steam as it leaves the first set of rotating buckets. O_1F, parallel to OA and equal in length to EC, represents the velocity of the steam as it enters the first stationary or reversing bucket. JG is the

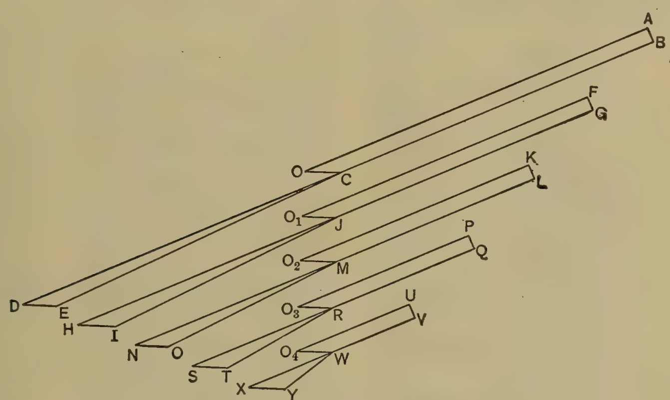


FIG. 247. Theoretical Velocity Diagram, Terry Turbine.

component of the resultant of O_1F and O_1J in line with the jet. The resultant IJ of HJ ($= JG$) and HI represents the velocity of the steam as it leaves the first stationary bucket. This construction is repeated through all velocity stages. The final exit velocity of the steam as it issues from the moving buckets is WY. The energy converted into useful work is

$$\frac{W}{2g} = (\overline{OA}^2 - \overline{WY}^2).$$

In the actual turbine friction losses would reduce the length of the velocity lines and increase the amount of energy rejected in the exhaust. The construction of the velocity diagram as modified by friction is similar to that described in paragraph 217, Fig. 240.

220. Kerr Turbine. — Fig. 248 shows a longitudinal section through an eight-stage Kerr steam turbine illustrating the *compound-pressure* or *multi-cellular* group of the impulse type. The rotor consists of a

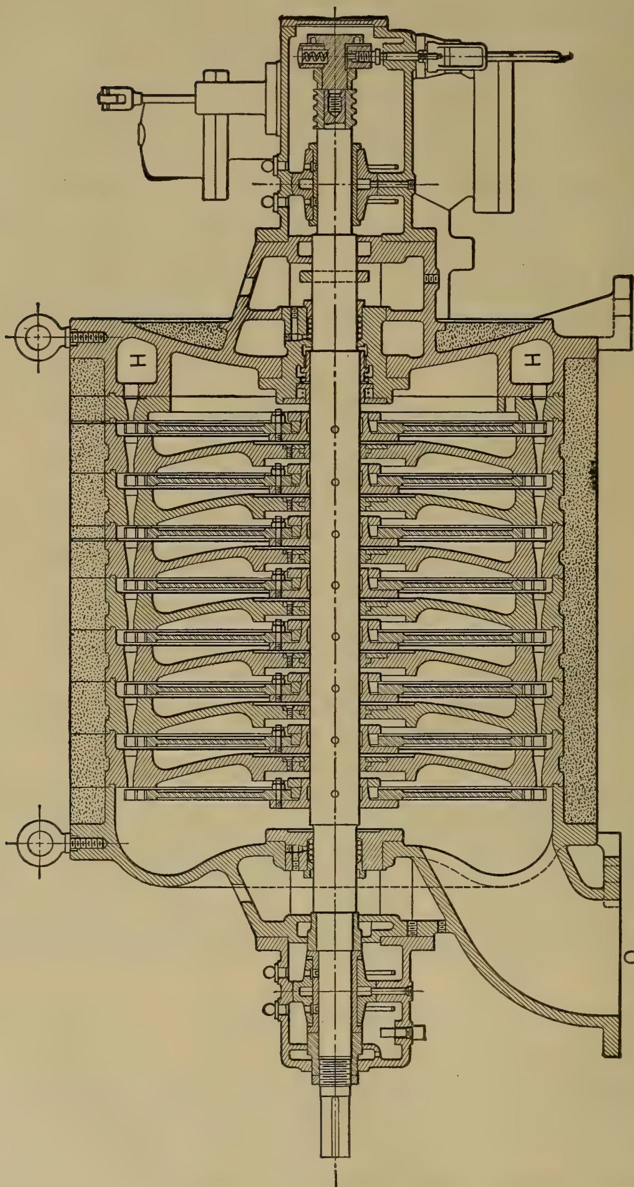


FIG. 248. Section through an Eight-stage Kerr Turbine.

series of steel disks, mounted on a rigid steel shaft. A series of drop-forged steel buckets is secured to the periphery and riveted in dove-tailed slots as shown in Fig. 249. The tips of the buckets are riveted to a shroud ring, thereby insuring a rigid and positive spaced construction. The stator is made up of a number of arched cast-iron diaphragms with circular rims tongued and grooved, and bolted to steam-end and exhaust-end castings. The nozzles are formed by walls within the diaphragm and thin Monel metal vanes die-pressed into shape and cast into the diaphragm. One set of nozzles and one wheel constitute a stage and the expansion is usually carried out in from six to ten stages, depending upon the condition of operation.

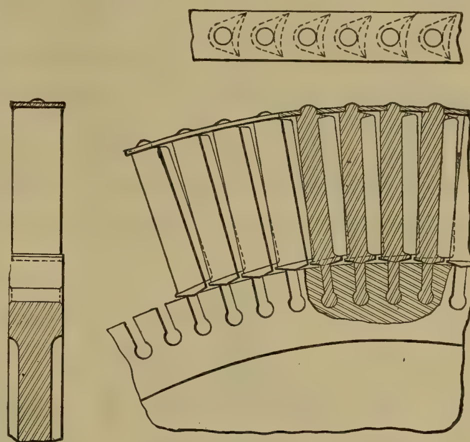


FIG. 249. Bucket Fastening, Kerr Turbine.

The operation is as follows: Steam enters the turbine through a double-beat balanced poppet valve, the stem of which is connected through levers to the governor, to the circular cored space H, H extending around the steam "end casting." This space acts as an equalizer and insures uniform admission to the first set of nozzles. Partial expansion takes place through the first set of nozzles and the kinetic energy is imparted to the rotor through the medium of the vanes. Steam leaves the buckets at a very low velocity and is again expanded through the second set of nozzles in the diaphragm. This process is repeated in each stage and exhaust steam leaves the turbine at O.

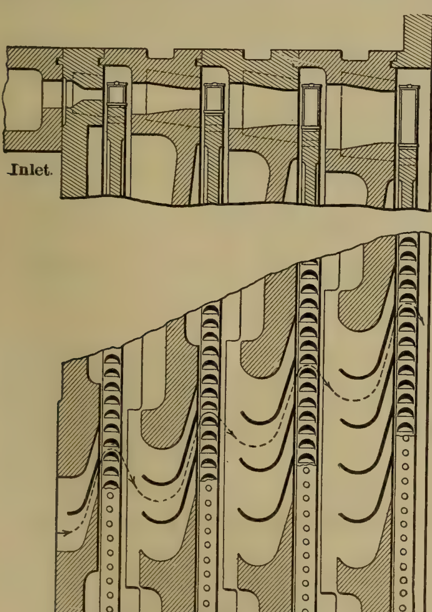


FIG. 250. Arrangement of Vanes and Nozzles, Kerr Turbine.

Fig. 251 illustrates the principles of the oil relay governor as applied to the larger sizes of turbines driving alternators. Referring to Fig. 251: rotation of the turbine shaft is transmitted through worm gear and governor spindle to weights, *W*, *W'*. Centrifugal force throws these weights outward about suspension points *A* and *A'*, overcoming the resistance of the spring. The movement of the spring is transmitted through lever *L* to relay plunger *P* and admits oil pressure (about 30 pounds per square inch) to piston *S* and in this manner throttles admission valve *V*. Similarly, a downward movement of the relay plunger stem releases oil pressure and opens the admission valve.

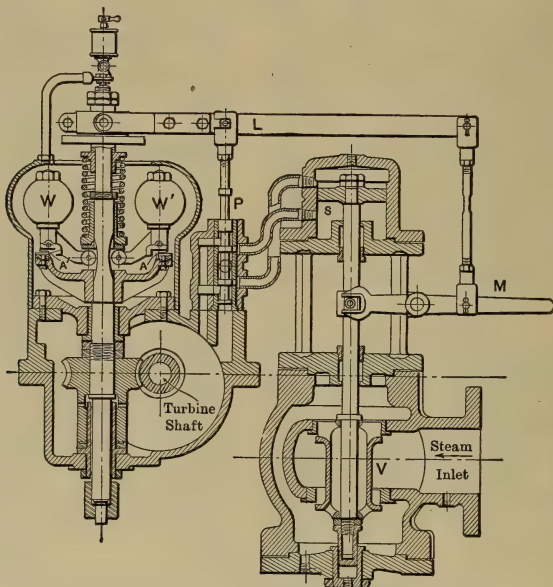


Fig. 251. Oil Relay Governor, Kerr Turbine.

Floating lever *L* is connected to the admission valve stem through secondary lever *M* so that the movement of the steam valve returns the relay plunger to its central position. This equalizes the pressure on top and bottom of the main piston *S* and arrests its movement, thereby maintaining a fixed opening for a given speed. A suitable emergency valve automatically cuts off the steam supply in case the speed exceeds a predetermined amount.

A spring-loaded governor of the centrifugal type mounted directly on the turbine shaft is used to control the smaller sizes of turbines.

Kerr turbines are constructed horizontally and vertically and in various sizes ranging from 5 to 750 horse power, and are designed to operate all classes of pumps, blowers and generators. The rotative speed

varies from 2000 to 4000 r.p.m., depending upon the service for which the turbines are intended.

221. The De Laval Multi-stage Turbine differs from the Kerr turbine only in mechanical details. It is of the multi-cellular type and is provided with a rigid shaft. The increase in the cross-sectional area of the passages required by the expansion of the steam as it proceeds through the turbine is effected by lengthening the blades, reducing the diameters of the wheels correspondingly and increasing the bore of the casing. (In the Kerr turbine the blades are lengthened and increased in width from the high-pressure to the low-pressure stages and the steam passages are increased in size but the outside diameter of the rotor remains the same.) The bearings are of rigid construction arranged for water cooling. Labyrinth packing is used between stages and combined labyrinth and carbon-ring packing at the steam and exhaust ends of the casing. Air leakage into the turbine is prevented by introducing live steam between the two outer carbon rings. The governor is of the throttling type and is mounted upon a vertical shaft driven through worm gearing by the main turbine shaft.

222. Elementary Theory. — Kerr Turbine. — In the Kerr turbine the diameters of the moving elements are uniform and hence the peripheral velocities are the same. In the frictionless turbine the velocity issuing from each jet or the stage velocity should be the same. If there are n stages the heat drop per stage will be $\frac{1}{n}$ of the total heat drop. Since there are no friction losses in the ideal turbine the total heat drop is

$$H_1 - H_n$$

and the heat drop per stage

$$\frac{H_1 - H_n}{n}.$$

The stage velocity or initial velocity of jet from each nozzle is

$$V = 224 \sqrt{\frac{H_1 - H_n}{n}}.$$

The pressure, specific volume and quality of the steam in each stage may be determined by subtracting $\frac{H_1 - H_n}{n}$ from the heat content of the preceding stage and finding the corresponding quantities from temperature-entropy tables or diagrams.

Thus, an eight-stage turbine operating non-condensing at 190 pounds initial absolute pressure would show about the following conditions.

(All friction and leakage losses neglected and final velocity in each stage assumed to be zero.)

$$H_1 = 1197.3 \text{ B.t.u. per pound.}$$

$$H_n = 1012.5 \text{ B.t.u. per pound.}$$

$$\text{Total heat drop} = H_1 - H_n = 1197.3 - 1012.5 = 184.8.$$

$$\text{Heat drop per stage} = \frac{184.8}{8} = 23.1$$

$$\text{Stage velocity} = 224\sqrt{23.1} = 1080 \text{ feet per second.}$$

Stage.	Heat Content.	Pressure, Lbs. Abs.	Quality, Per Cent.	Specific Volume Cu. Ft. per Lb.
Admission.	1197.3	190	100	2.41
1	1174.2	145	97.9	3.04
2	1151.1	109	95.9	3.93
3	1127.0	80	94.0	5.14
4	1104.9	58	92.2	6.77
5	1081.8	42	89.6	8.96
6	1058.7	30	88.8	12.07
7	1035.6	21	87.3	16.33
8	1012.5	Atmospheric	85.8	22.55

In the actual turbine only 50 to 75 per cent of the heat theoretically available is transformed into useful work. A small portion is lost by gland leakage, radiation and bearing friction and the balance has been retransformed from kinetic energy into potential energy by eddying, fluid friction and blade leakage. The efficiency of each stage is less than that of the turbine as a whole since the increase in heat content due to friction, etc., is available for transformation into useful work in the succeeding stages. To find the actual pressure condition in each stage allowing for the various losses, it is necessary to correct the theoretical quantities for these losses. See "Energy and Pressure Drop in Compound Steam Turbines," by F. E. Cardullo, Proc. A.S.M.E., Feb., 1911, and paper read by Prof. C. H. Peabody, Proc. Society of Naval Architects and Marine Engineers, June, 1909.

223. The Curtis Steam Turbine. — Figs. 252 to 265 show the general arrangement and a few details of the Curtis steam turbine, which is of the compound pressure and velocity type. The total expansion is carried out in one or more compartments or stages, each stage comprising a set of expanding nozzles and a wheel carrying two or more rows of buckets. A high initial velocity is given to the jet in each stage by expansion in the nozzles as in the De Laval, and the energy is absorbed by successive action upon the series of moving and stationary vanes

arranged somewhat as in the Parsons turbines, paragraph 226. In the latter, however, the difference in pressure between the two sides of each vane induces flow by continuous expansion, while in the former the moving vanes in any one stage simply absorb the kinetic energy already created by expansion in the nozzle. The action is as follows: Steam enters stage (1), Fig. 252, through the first set of nozzles, and is partially expanded. With the resulting initial velocity it impinges against the first row of moving blades and gives up part of its energy, and is then deflected through the adjoining stationary blades to the next set of moving vanes, where its velocity is still further reduced, and

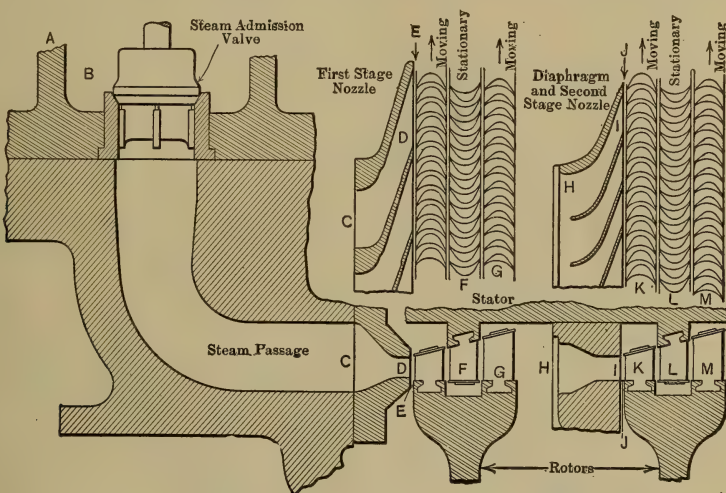


FIG. 252. Arrangement of Nozzles and Blades, Curtis Turbine.

so on until it has been brought practically to rest. From this stage the steam flows at reduced pressure through nozzles of second stage, which are sufficient in number and in size to afford the greater area required by the increased volume. In expanding in these nozzles it acquires new velocity and gives up energy to the moving blades as before. This process is repeated through two to five stages, depending upon the size of turbine. Fig. 253 shows a partial section of a four-stage vertical 5000-kw. machine. *R, R* are sections through the revolving wheels, which in this particular turbine are nine feet in diameter and keyed to the vertical shaft *S*. On the periphery of each wheel are bolted two rows of blades or vanes, with a stationary or intermediate row attached to the casing between them. The buckets are made of rolled nickel bronze, hammered to shape and finish. The roots are dovetailed into the holders and the tips are tenoned and riveted into a

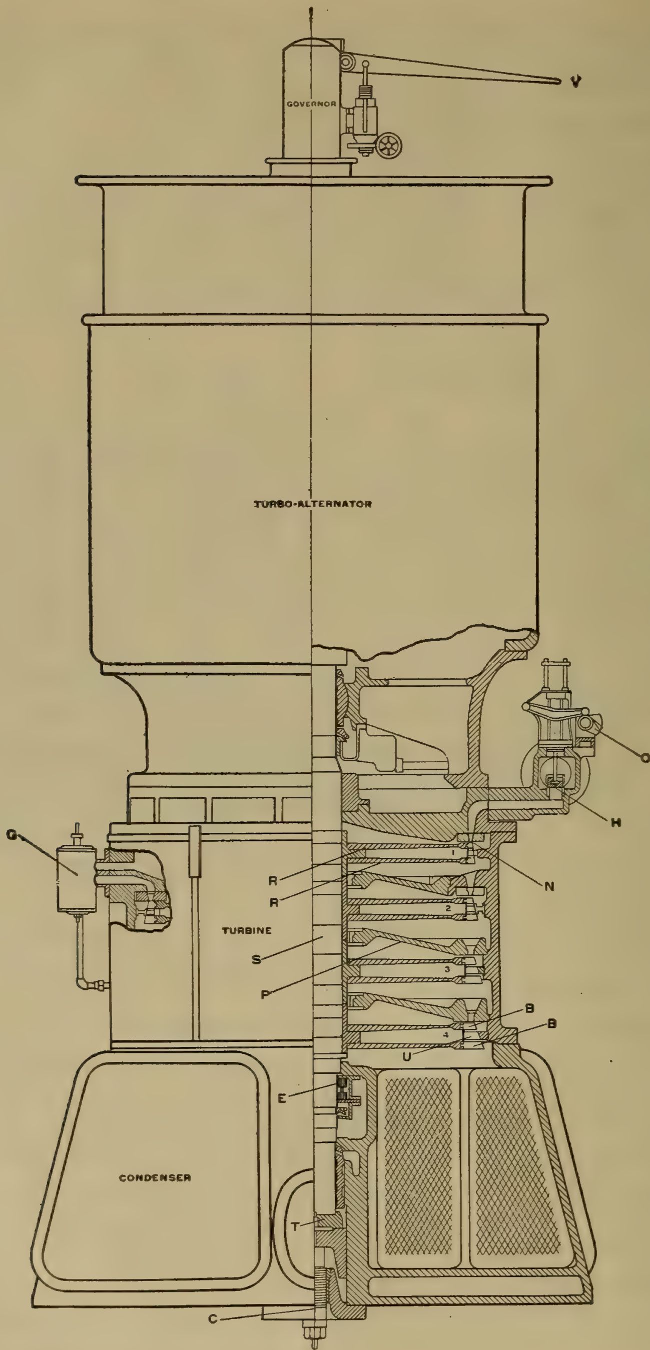


FIG. 253. Four-stage Vertical Curtis Turbo-Generator. Base Condenser Type.

shroud ring, thus insuring positive spacing and a rigid construction. See Fig. 254. Between each pair of wheels is a stationary steam-tight diaphragm P, which contains the nozzles through which the steam is expanded from the preceding stage. It will be noticed that the buckets and nozzles increase rapidly in size in succeeding stages as the pressure falls and the volume of steam increases. The parts are so proportioned that the steam gives up approximately $\frac{1}{n}$ of its energy in each stage, n representing the number of stages. The number of stages and the number of vanes in a stage are governed by the degree of expansion, the peripheral velocity which is desirable or practicable, and by various conditions of mechanical expediency. The admission valves vary in number and in location with the size of turbine. The automatic stage valve G connects the first stage directly to a set of auxiliary second-stage nozzles. See Fig. 255. Thus the overload capacity is increased by widening the steam belt and not by admitting high-pressure steam into an intermediate stage as was formerly the practice with Curtis turbines. This method of overload control results in higher efficiency than the older system.

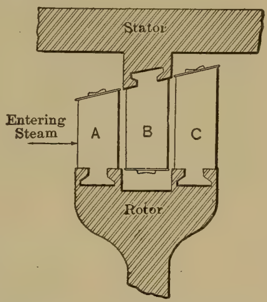


FIG. 254. Bucket Details, Curtis Turbine.

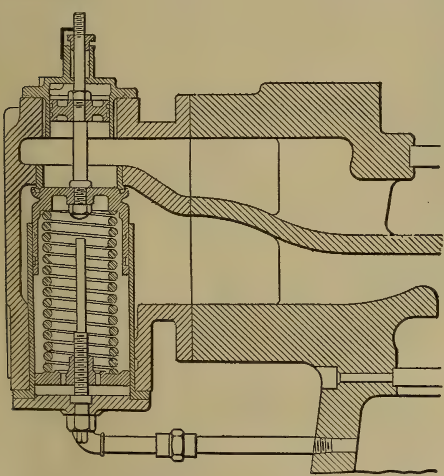


FIG. 255. Automatic Stage Valve, Curtis Turbine.

Curtis turbines appear to have a wider range of economical application than any other type, commercial sizes ranging from a small horizontal unit of 7 kilowatts rated output to vertical units of 25,000 kilowatts capacity on the continuous 24-hour basis. The smaller machines, 7500 kilowatts and under, are usually of the horizontal type, and the larger units, 9000 kilowatts and larger, are of the vertical type. All Curtis turbines are governed by "cutting-out nozzles"; that is, full initial pressure is maintained in all the nozzles that are open and the capacity of the machine is controlled by varying the number in operation. Units under 500 kilowatts are ordinarily controlled by a

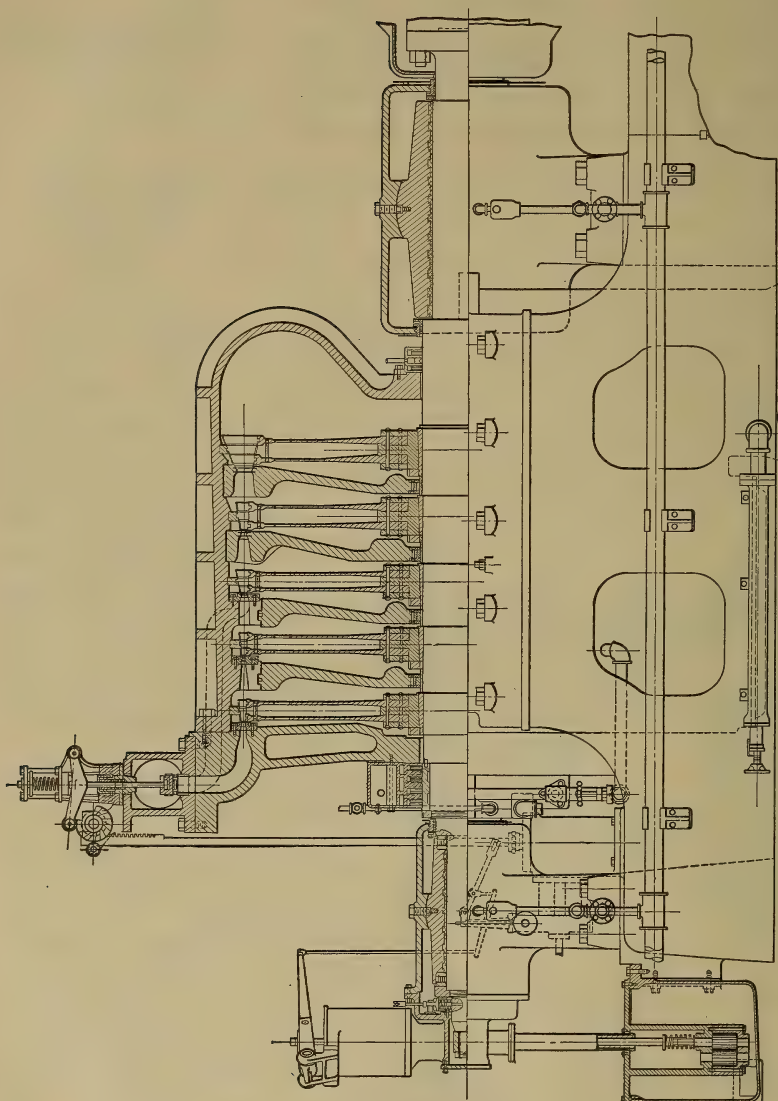


FIG. 256. Longitudinal Section through a 3500-Kva. Horizontal Curtis Steam Turbine.

mechanical valve gear and the larger units by an *indirect* or *relay* system. In the older types this relay system was electrically operated; in the modern machines the valves are hydraulically controlled.

Fig. 257 shows a section through the governor for the large units. Speed regulation is accomplished by the balance maintained between the centrifugal force of moving weights *AA* and the static force exerted by spring *D*. The governor is provided with an auxiliary spring *F* for varying its speed when synchronizing, the tension in which is varied by a small pilot motor controlled from the switchboard. The movement

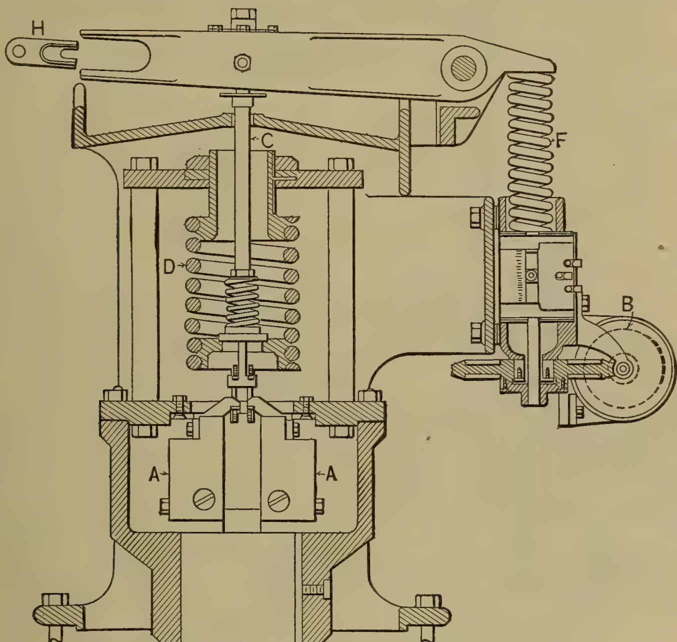


Fig. 257. Main Governor, Curtis Turbine.

of the governor weights is transmitted through rod *C* to arm *H* and by means of the latter to the controlling mechanism of the valve gear.

Fig. 258 gives an assembly view of the mechanical valve gear as applied to a 300-kilowatt unit. The valve stems extend upward through ordinary stuffing boxes and are attached to notched crossheads 8, 8. Each crosshead is actuated by a pair of reciprocating pawls or dogs, 6, 6, the lower one of which closes the valve and the upper one opens it. The several pairs of pawls are hung on a common shaft which receives a rocking motion from a crank driven by the turbine shaft. The crossheads have notches milled in the side in which the pawls engage to open or close the valve, the engagement being determined by shield

plates 2, the positions of which are controlled by the governor through the medium of suitable levers. Shield plates 6 are set one a little ahead of the other to obtain successive opening or closing of the valves. The pawls are held in position when not in contact with the shield plates by springs *W*.

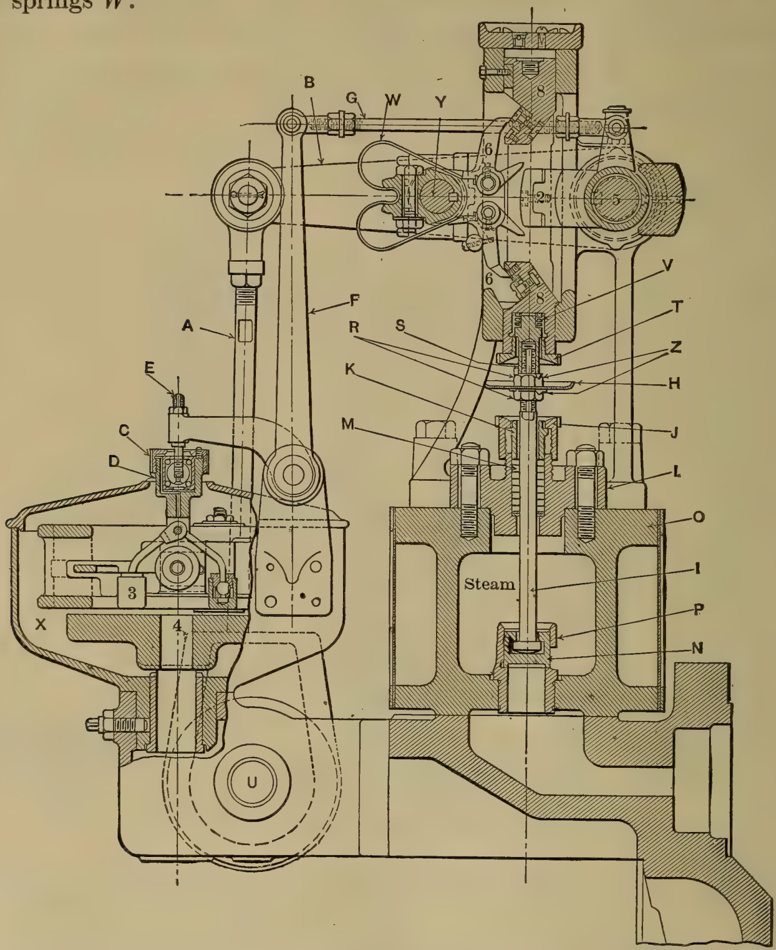


Fig. 258. Assembly of Mechanical Valve Gears for 300-Kw. Curtis Steam Turbine.

Fig. 259 gives a diagrammatic arrangement of the hydraulically controlled valve-gear mechanism. The motion of governor *g* is transmitted through lever *i* to lever *a* of the pilot valve *j*. Pilot valve *j* controls the supply of oil (under pressure) in cylinder *k*, the piston of which actuates rods *l*, *l*.* The movement of rod *l* is transmitted through

* The pressure is on both sides of the piston and the latter is actuated by releasing the pressure.

rack m to a small pinion. This pinion is mounted on the end of a shaft fitted with a number of cams, one a little ahead of the other, each cam controlling the opening and closing of a steam valve through the medium of rocker arm f. As the load on the turbine increases the governor slows down and causes the cam shaft to rotate in a reverse direction, in-

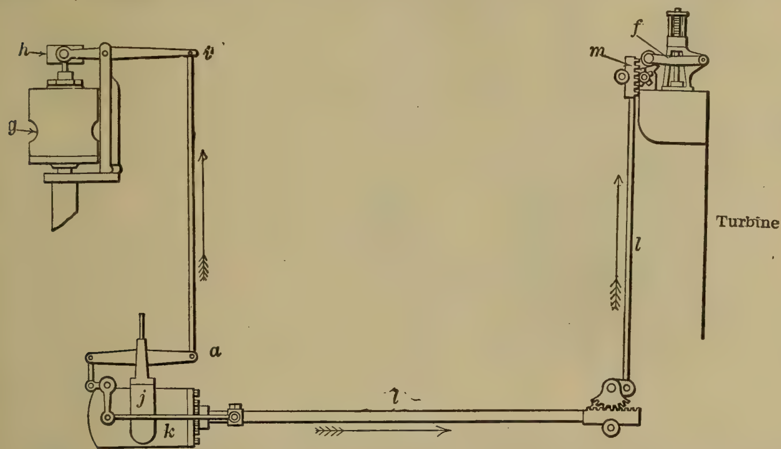


FIG. 259. Diagrammatic Arrangement of Hydraulically Operated Valve Gear, Curtis Turbine.

dicated by the arrow points in Fig. 259. This causes a proportionate number of valves to be lifted and held open, the number increasing as the load increases, until all are open. Should the load continue to increase, as in the case of overload, the secondary valve opens as previously described, connecting the first stage with a set of auxiliary second-stage nozzles. Only the nozzles in the first stage are controlled by the

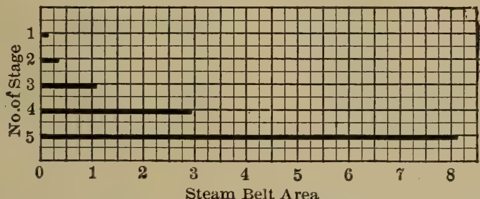


FIG. 260. Steam-belt Area in Five-stage Curtis Turbine.

governor. Should the turbine run above normal speed the emergency stop-valve automatically closes the admission of steam to the nozzles. This device consists of a steel ring placed around the shaft between the turbine and the generator. This ring is eccentrically mounted and the unbalanced centrifugal force is balanced by a helical spring. When the predetermined speed is reached the centrifugal force overcomes the

spring tension and the ring moves in a still more eccentric position. In this position the ring strikes a bell-crank lever which trips the throttle valve and permits it to close by its own weight and the unbalanced pressure on the valve stem.

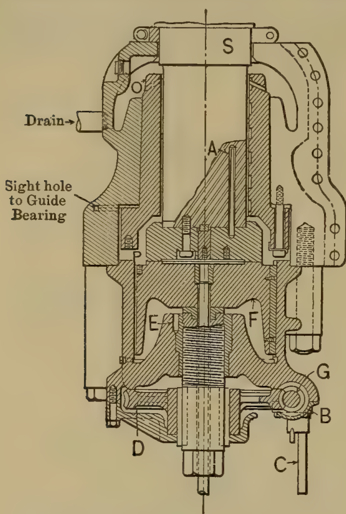


Fig. 261. Step Bearing, Vertical Curtis Turbine.

In the Curtis turbine the area of the steam admission is limited to a small portion of the circumference in the first stages and does not extend around the entire circumference until the last stage is reached. See Fig. 260.

The step bearing of a vertical machine is illustrated in Fig. 261. The weight of the rotor is supported by oil under pressure forced between the bearing blocks *F* and *P*, thus permitting the shaft *S* to revolve on a film of oil. The smaller disk *P* is attached by dowels to the main shaft. Carbon packing rings are used above the bearing to prevent leakage, and adjustment is provided in the lower bearing block by means of worm gear *D*, worm *G* and ratchet handle *C*. The oil pressure varies from 150 to 750 pounds per square inch according to the size of machine, the higher pressures being used in the larger machines. The carbon packing and casing for the high-pressure end of the vertical turbine is shown in Fig. 262. Figs. 263 and 264 give general details of the carbon packing and water-cooled bearings of the horizontal turbine.

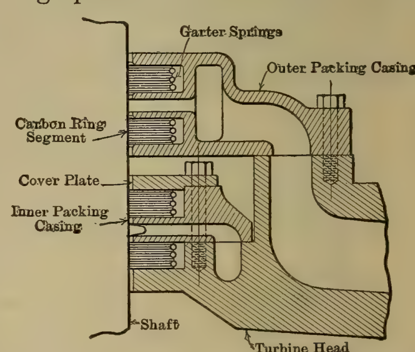


Fig. 262. Carbon Packing, Vertical Curtis Turbine.

Fig. 265 gives a diagrammatic outline of the oiling system. A tank, of sufficient capacity to contain all the oil and fitted with suitable straining devices and a cooling coil, is located at a level low enough to receive oil by gravity from all points lubricated. A pump draws oil from this tank and delivers it at a pressure about 25 per cent higher than that required to sustain the weight of the turbine in the step bearing. A spiral duct baffle connects the source of pressure to the step bearing and serves to regulate the oil supply to the lower end of

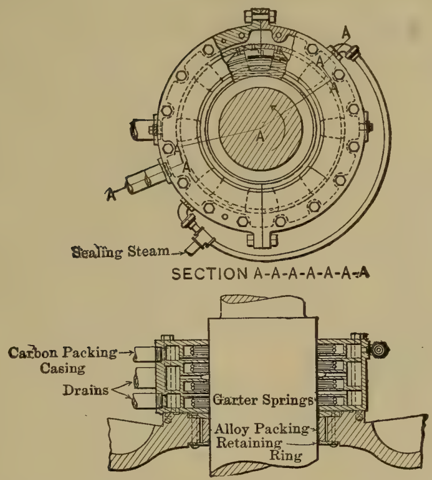


FIG. 263. Carbon Packing, Horizontal Curtis Turbine.

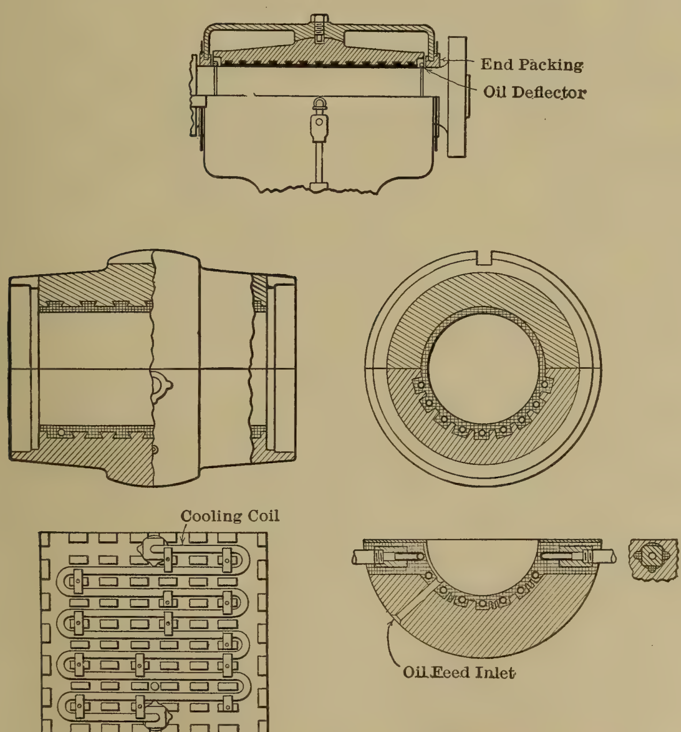


FIG. 264. Water-cooled Bearing, Horizontal Curtis Turbine.

the shaft. This source of pressure is also connected through a reducing valve to the upper oiling system of the machine, in which a pressure of about 60 pounds to the square inch is maintained. This system, which includes a storage tank partly filled with compressed air, operates the hydraulic governor mechanism and supplies oil to the upper bearings. Delivery of oil to these bearings is regulated by adjustable baffles designed to offer resistance to the oil flow without forcing the oil to pass through any very small opening which might easily become clogged. A relief valve is provided to prevent the pressure in the upper part of the oiling system from rising above a desirable limit. Drain pipes from the upper bearings and from the hydraulic cylinder and relief valve all discharge into a common chamber, in which the streams are visible, so that the oil distribution can always be easily observed. At some point in the high-pressure system adjacent to the pump it is desirable to install a device to equalize the delivery of oil from the pump, as is done by the air chamber commonly used with pumps designed for low pressure. A small spring accumulator is furnished for this purpose, except in cases where weighted storage accumulators are used. In large stations where several machines are installed, a storage accumulator is desirable and can be arranged advantageously so that it will normally remain full, but will discharge if pressure fails, and in doing so will start auxiliary pumping apparatus.

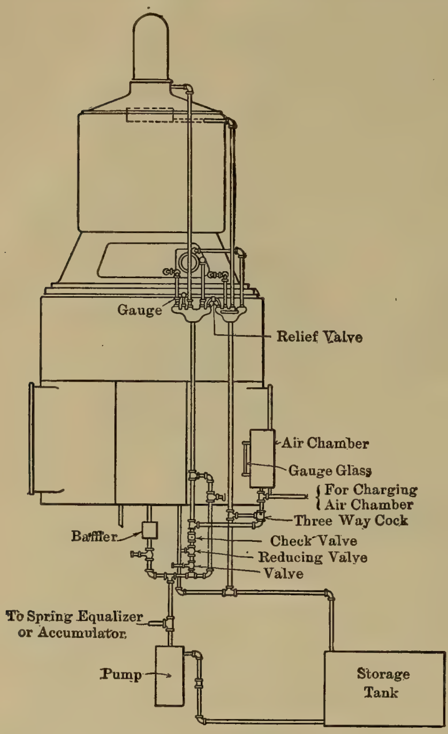


FIG. 265. Arrangement of Oiling System for Curtis Turbine.

DIRECT CURRENT.

Kw.	R.p.m.	Kw.	R.p.m.
15	4,000	150	2,000
25	3,600	300	1,800
75	2,400	500	1,500

ALTERNATING CURRENT.

300	1,800	2,000	900
500	1,800	3,000	600–750
1,000	1,200	to	
1,500	900	20,000	

For the description of a typical steam-turbine station equipped with Curtis turbines see Chapter XX.

General Description of Curtis Turbines: Power, March, 1909; Engr. U. S., Jan. 1, 1908, p. 115; Power & Engr., Feb. 25, 1908, p. 284, Feb. 25, 1908, March 3, 1908; Elec. Wld., June 17, 1905, p. 1136.

Guide Bearings, Oil Distribution & Carbon Packing: Power & Engr., April 14, 1908; Jan. 3, 1911, p. 10.

Mechanical Valve Gear: Power & Engr., March 10, 1908, p. 356.

Hydraulic Valve Gear: Power, March, 1909, p. 189.

Heat Losses: Power & Engr., Jan. 3, 1911, p. 19; Proc. A.S.M.E., Feb., 1911, p. 123.

224. Elementary Theory, Curtis Turbine. — Fig. 266 gives a diagrammatic arrangement of the blades and nozzles in the first stage of a two-stage Curtis turbine, each stage consisting of one set of nozzles and two moving and one stationary sets of blades.

Referring to the diagram: the steam is expanded in the first stage from pressure P_1 to P_2 and issues from the first set of nozzles with *absolute* velocity V_1, striking the first set of moving blades at an angle α with the line of motion of the wheel. The resultant v_1 of V_1 and the peripheral velocity u is the velocity of the steam *relative* to the vanes; and the angle β which the line v_1 makes with the line of motion of the wheel is the proper entrance angle of the blades for the first set. Neglecting friction the exit angle γ will be the same as the entrance angle β. The resultant of v_2, the exit velocity *relative* to the blade, and u, the peripheral velocity, is V_2, the *absolute* exit velocity.

Since the second set of blades is fixed and serves as a means of changing the direction of flow, the absolute velocity entering them is V_2. The angle δ formed by V_2 and the center line of the stationary blades is the proper entrance angle. Neglecting friction the absolute exit velocity will be $V_3 = V_2$, and the exit angle will be $\epsilon = \delta$. The steam flowing from the stationary blades strikes the second set of moving blades at an angle $\epsilon = \delta$ with *absolute* velocity V_3. Combining V_3 with the peripheral velocity u we get v_3, the velocity of the steam *relative* to the second set of moving blades. The angle θ, formed by v_3 and the line of motion of the wheel, is the proper entrance angle for the second set of moving blades. The resultant of v_4 $(= v_3)$ and u is V_4, the absolute exit velocity for the first stage.

In the second stage the steam is expanded from pressure P_2 to that in the condenser and acquires initial velocity V_a, leaving the last bucket with residual velocity V_n. The theoretical velocities and blade angles for this stage may be found as above.

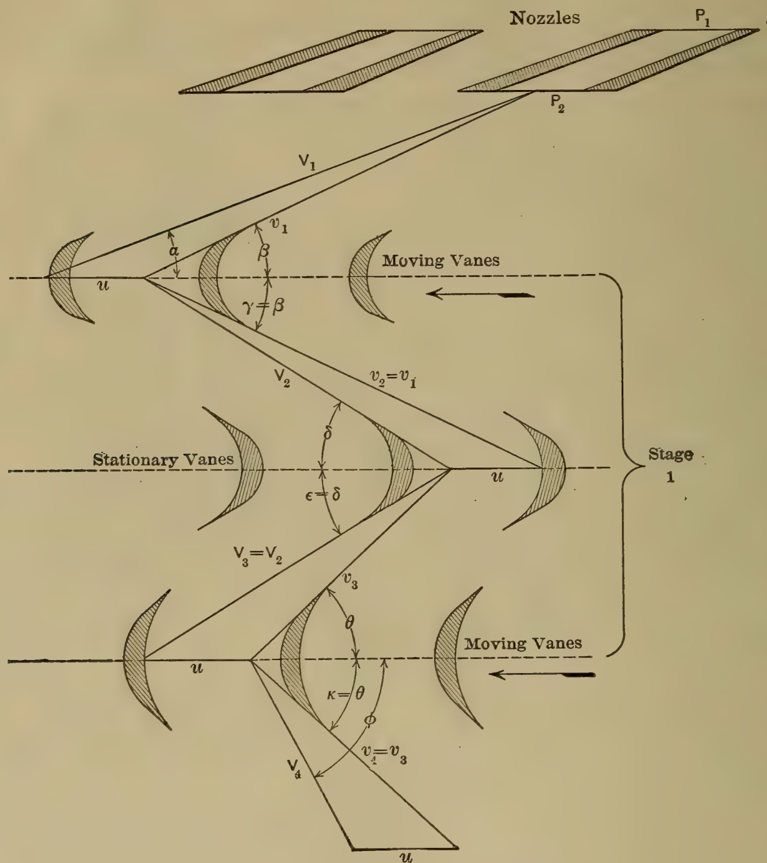


FIG. 266. Velocity Diagram, Curtis Turbine.

Example: A four-stage Curtis turbine develops 800 horse power on a steam consumption of 12 pounds per horse-power hour; initial pressure 150 pounds absolute, superheat 100 degrees F., back pressure 1.5 pounds absolute, peripheral velocity 450 feet per second, angle of the nozzle with the plane of rotation, 20 degrees. Each stage consists of two rotating elements and one stationary element. Compare the performance of the actual turbine with its theoretical possibilities.

Ideal Turbine:

For the sake of simplicity it will be assumed that the final velocity of each stage is zero and that the heat drop in the first set of nozzles is one fourth of the total theoretical drop assuming adiabatic expansion.

From steam tables $H_1 = 1249.6$ B.t.u.

From entropy tables or Mollier diagram $H_n = 934.6$.

Total heat drop $= 1249.6 - 934.6 = 315$.

Heat drop in first stage $\frac{315}{4} = 78.75$.

The velocity of the jet in the first stage is

$$V_1 = 224\sqrt{78.75} = 1985 \text{ feet per second.}$$

By laying off this initial velocity in direction and amount and combining it with the peripheral velocity as in Fig. 266, the absolute velocities V_2 and V_3 may be readily obtained.

The kinetic energy absorbed in the first set of moving blades, per pound of steam, is

$$E_1 = \frac{1}{64.4}(V_1^2 - V_2^2)$$

$$= \frac{1}{64.4}(1985^2 - 1170^2) = 39{,}930 \text{ foot pounds per second,}$$

and in the second set of moving blades

$$E_2 = \frac{1}{64.4}(V_2^2 - V_3^2)$$

$$= \frac{1}{64.4}(1170^2 - 670^2)$$

$$= 14{,}280 \text{ foot-pounds per second.}$$

The total energy converted into useful work is

$$39{,}930 + 14{,}280 = 54{,}200 \text{ foot-pounds per second.}$$

Had the entire heat drop been utilized in doing work the total energy would be

$$\frac{1}{64.4} \times 1985^2 = 61{,}180 \text{ foot-pounds per second.}$$

The difference $61{,}180 - 54{,}210 = 6970$ represents the loss due to the residual velocity of the steam leaving the last bucket.

Since the steam is brought to rest before entering the second set of nozzles, the heat equivalent of this energy or $\frac{6970}{778} = 8.96$ B.t.u. increases the final heat content; thus

$$H_2 = 1249.6 - 78.75 + 8.96 = 1179.8 \text{ B.t.u.}$$

But a total heat drop per stage of 78.75 B.t.u. was assumed as a requirement of the problem and the final result obtained above shows it to be 78.5 − 8.96 = 69.54. By trial and adjustment or by means of empirical formulas a value of H_2 may be obtained which will fulfil the given conditions. Such an analysis is beyond the scope of this book, and the reader is referred to Forrest E. Cardullo's article "Energy and Pressure Drops in Compound Steam Turbines," Trans. A.S.M.E., vol. 33, p. 325, 1911.

The remaining stages may be analyzed in a similar manner.

It should be borne in mind that in the actual turbine the velocity will be less than the theoretical on account of frictional resistances in the nozzles and blades and the heat content H_1, H_2 . . . H_n will be greater than that of the ideal mechanism. Radiation, leakage, windage and other losses must also be considered in determining actual conditions.

Neglecting the residual energy in the exhaust, the total heat drop $H_1 - H_n$ is available for doing useful work and the water rate of the ideal turbine is

$$W = \frac{2546}{H_1 - H_n} = \frac{2546}{315} = 8.1 \text{ pounds per horse-power hour.}$$

Heat consumption per horse-power hour per minute

$$= \frac{8.1(1249.6 - 83.9)}{60} = 157 \text{ B.t.u.}$$

Thermal efficiency

$$E_r = \frac{1249.6 - 934.6}{1249.6 - 83.9} = 0.27.$$

Actual Turbine:

Steam used per hour = 800 × 12 = 9600 pounds.

Steam used per second = 9600 ÷ 3600 = 2.66 pounds.

Horse power developed per pound of steam flowing per second = 800 ÷ 2.66 = 300.

Kinetic energy converted into useful work

300 × 550 = 165,000 foot-pounds per second.

Thermal efficiency, equation (138)

$$E_t = \frac{2546}{12\,(1249.6 - 83.9)} = 0.182.$$

Heat consumption, B.t.u. per horse power per minute,

$$\frac{12\,(1249.6 - 83.9)}{60} = 233.$$

$$\text{Efficiency ratio} = \frac{E_t}{E_r} = \frac{0.182}{0.270} = 0.675.$$

SUMMARY

	Actual Turbine.	Perfect Turbine.
Horse power developed per pound of steam	300	445
Steam consumption, pounds per horse-power hour	12.0	8.1
B.t.u. consumed per horse power per minute	233	157
Thermal efficiency, per cent	18.2	27.0
Efficiency ratio, per cent	67.5	....

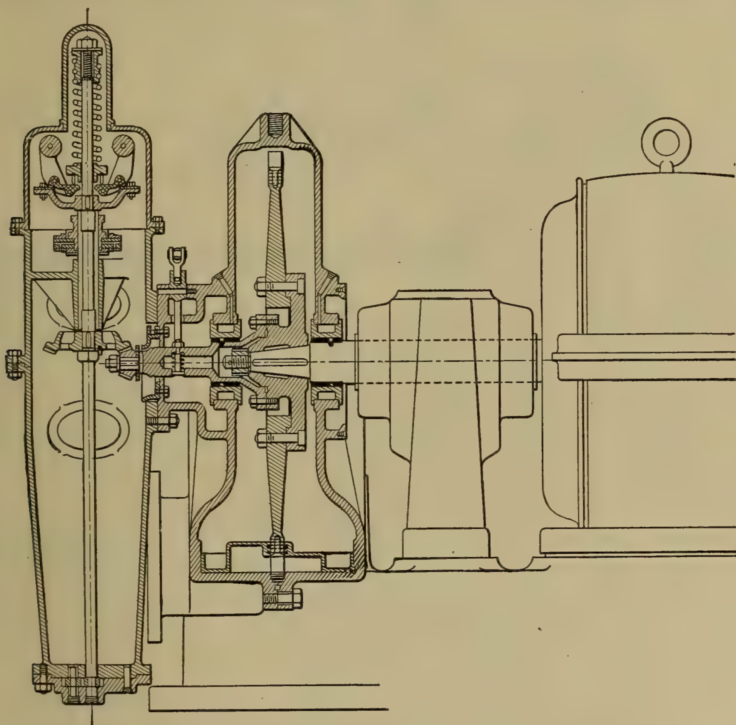


Fig. 267. Section through Westinghouse Impulse Turbine.

224. Westinghouse Impulse Turbine. — To meet the demand for small turbines the Westinghouse Machine Company has recently placed on the market turbines ranging in size from 25 to 500 horse power. Units under 50 horse power are of the single-stage multi-velocity type and the larger sizes are of the two-stage multi-velocity type. All of these turbines have but one moving element. Fig. 267 shows a section through one of the larger units and Fig. 268 a diagrammatic arrangement of blades and nozzles. Referring to Fig. 268, steam enters the turbine at A, through a double-seated poppet valve, and is partially

expanded in nozzle B. Leaving the nozzle it impinges on the moving blades, giving up part of its energy, and thence passes to the first reversing chamber C, where its direction is changed and it is redirected against the moving blades. The steam is then expanded through the second set of nozzles to the existing back pressure and the cycle is repeated. Turbines of 50 horse power or less have but one set of nozzles and one reversing chamber, all others having two sets of nozzles and reversing chambers.

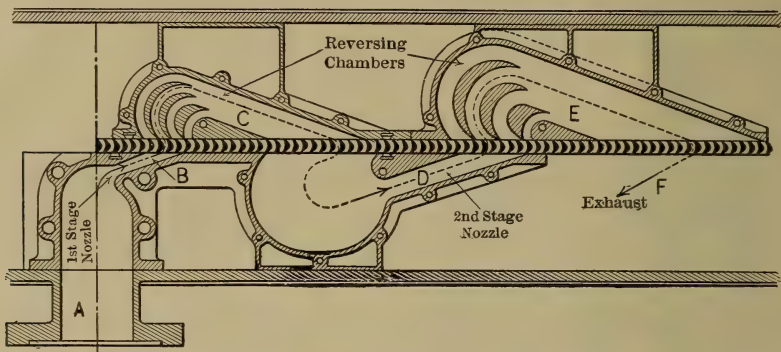


Fig. 268. Arrangement of Blades and Nozzles — Westinghouse Impulse Turbine.

226. Westinghouse Single-flow Steam Turbine. — Fig. 269 shows a section through a Westinghouse turbine of the original Parsons design and illustrates the multi-stage reaction type. In this type no nozzles are employed and expansion of the steam is effected by a series of stationary and moving blades. The rotor is a steel barrel or drum divided into three sections of varying diameter, upon the periphery of which bronze blades are radially inserted in dovetailed grooves. The adoption of three sections of varying diameter has no bearing on the design of this machine but is merely for mechanical convenience. The blades increase in length and cross section from the high-pressure to the low-pressure end of each section. The stator is of cast iron and its inner surface is studded with rows of blades projecting radially inward and conforming in size with the adjoining blades of the rotor. The relative positions of the blades in the rotor and stator are shown in Fig. 270. The operation of the turbine is as follows: Steam enters at S, Fig. 269, through poppet valve V, which is actuated by the governor shown in detail in Fig. 273, and flows through the annular space between rotor and stator to the exhaust opening at B. The entire expansion is carried out within this annular compartment and resembles in effect a simple divergent nozzle with the exception that the dynamic relationship of jet and vane is such as to secure a comparatively low velocity from inlet to exhaust. The

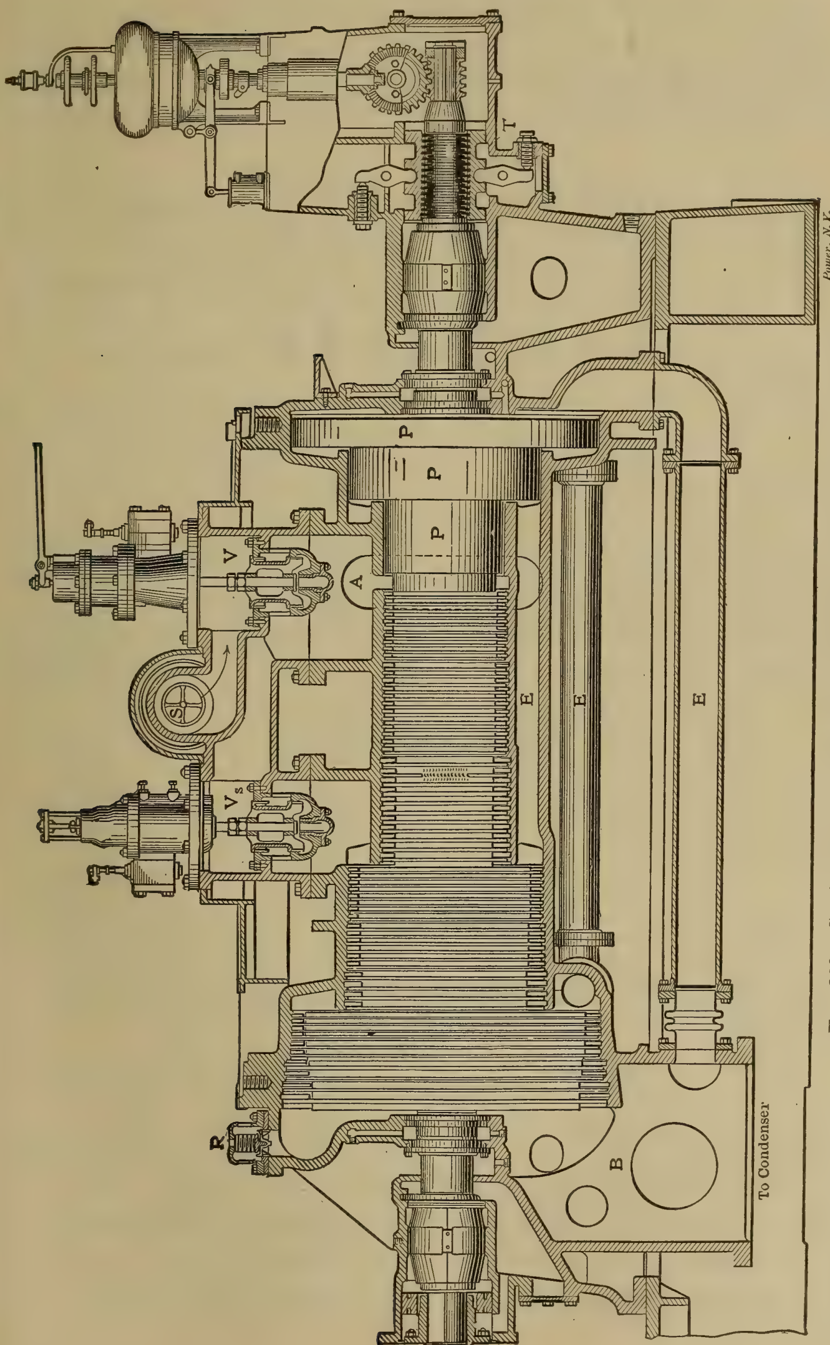


FIG. 269. Section through Westinghouse-Parsons Standard Steam Turbine.

velocity varies from 150 feet per second at the high-pressure end to about 600 feet per second as a maximum at the low-pressure end. The action of the steam on the blades is illustrated in Fig. 270. The steam strikes the first set of stationary blades as at P with initial velocity of about 150

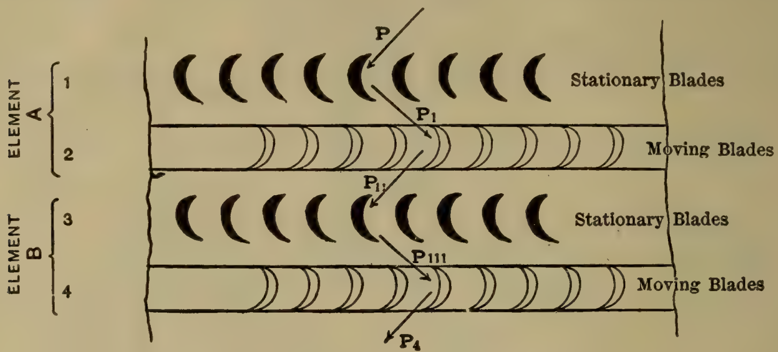


FIG. 270. Flow of Steam in Parsons Turbine.

feet per second and is deflected against the moving blades immediately adjoining. In passing from P to P_1 the steam is partly expanded and gives up a portion of its energy to the moving blades. The steam is deflected from P_1 to P_{11} and thus has a reactive effect on the moving blades in addition to the impulse imparted at P_1. The total torque produced

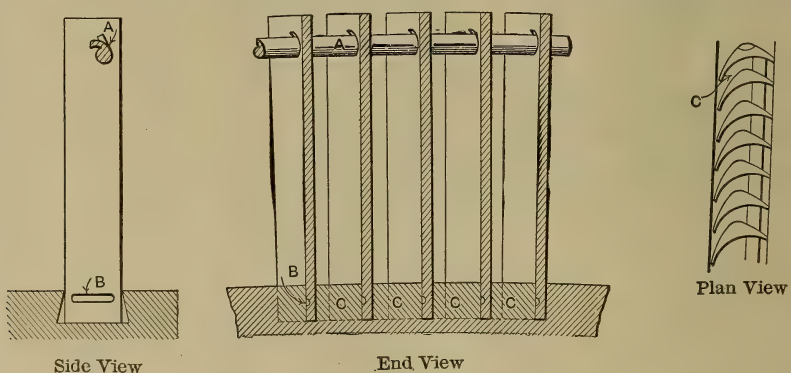


FIG. 271. Old Method of Fastening Blades in Westinghouse-Parsons Turbines.*

at the shaft in element A is therefore due to impulse from 1 and reaction from 2. This process is repeated in each element of the turbine, the steam expanding as it flows from element to element in its passage to the condenser. Opposed to the three sets of blades the spindle also carries three rotating balance pistons P, P, Fig. 269, each of such diameter as to ex-

* For present construction of interlocking root of blade see Jour. A.S.M.E., Aug., 1912, p. 1157.

actly balance, through passage E, the axial thrust of the steam against its corresponding drum of blades.

Steam enters the turbine intermittently as shown in Fig. 272, which represents indicator cards from a 1250-kilowatt turbine at various loads.

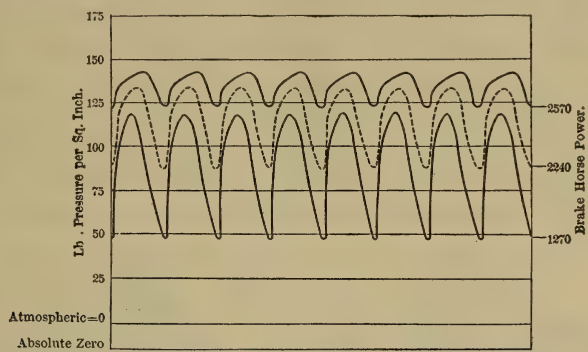


Fig. 272. Indicator Cards Showing Initial Pressure in a Westinghouse-Parsons Steam Turbine.

At light load the valve opens for a very short period. As the load increases, the period lengthens until finally, at about full load, the valve does not reach its seat at all, and continuous pressure is obtained in the high-pressure end of the turbine.

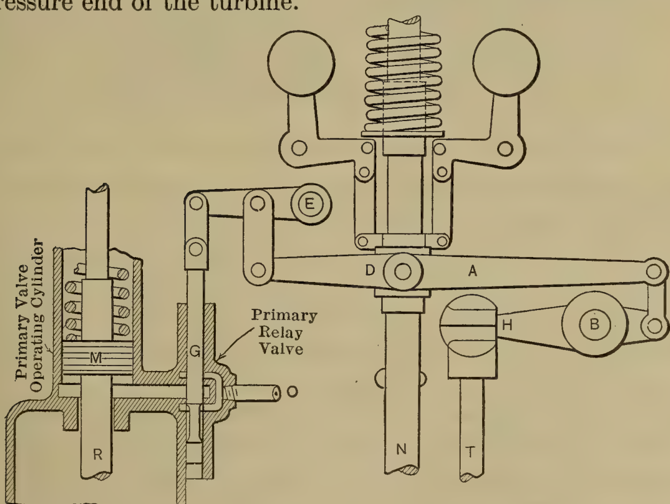


Fig. 273. Governor Mechanism, Westinghouse-Parsons Turbine.

The intermittent admission of steam is produced and controlled as follows: Lever T, Fig. 273, is given a reciprocating motion by an eccentric actuated by a worm and worm wheel on the main shaft. This motion is transmitted through lever H (with fixed fulcrum B) to lever A

(with floating fulcrum D) and finally to pilot valve G. This reciprocating pilot valve admits puffs of steam from pipe O to the under side of piston M, the rod R of which is attached to the admission valve V in Fig. 269. A spiral spring holds piston M in its lowest position until steam admitted by the pilot overcomes the spring tension and lifts the main valve from its seat, thereby permitting steam to enter the turbine. The fulcrum D of lever A is raised and lowered by the governor and therefore the pilot valve is controlled both by the motion of the eccentric and the motion of the governor. The eccentric keeps the pilot valve, and hence the main throttle, in constant oscillation, while the movement of the governor changes the limits of this motion.

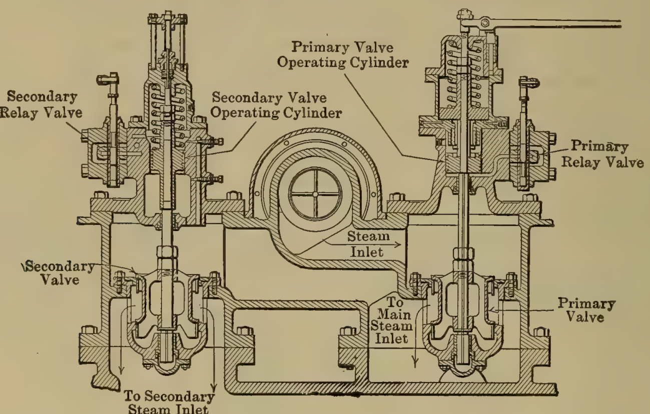


FIG. 274. Valve Gear with Steam Relay, Westinghouse-Parsons Turbine.

If an overload is sufficiently great to cause the governor balls to drop to their lowest position, the auxiliary or secondary valve V_s, Fig. 269, begins to open and admits high-pressure steam to the later stage where the working steam areas are greater, thus increasing in proportion the total power of the turbine. The operation of this valve is the same as the main admission valve and is controlled by the governor. Fig. 274 shows the details of this mechanism.

Fig. 275 shows the general details of the valve gear for the large units. Relay valve A is controlled by the governor and admits pressure oil to or exhausts it from the operating cylinders. When oil is admitted to the operating cylinder raising the piston, the lever C lifts the primary valve E. The lever D moves simultaneously with C, but on account of the slotted connection with the stem of the secondary valve F, the latter does not begin to lift until the primary valve is raised to the point at which its effective opening ceases to be increased by further upward travel.

In the smaller-sized machines running above 1200 r.p.m. flexible bearings are employed to absorb the vibration incident to the *critical* velocity. They consist of a nest of loosely fitting concentric bronze sleeves with sufficient clearance between them to insure the formation of a film of oil. In the larger machines a split self-aligning bearing is used instead of the flexible bearing. The ends of the casing are fitted with a water-seal made by a revolving wheel to prevent the escape of steam or inflow of air at the point of entry of the shaft. Leakage between dummy or balance cylinders is prevented by labyrinth packing which consists essentially of a series of grooves cut into the rotor into which bronze strips project from the casing. The steam in passing through the

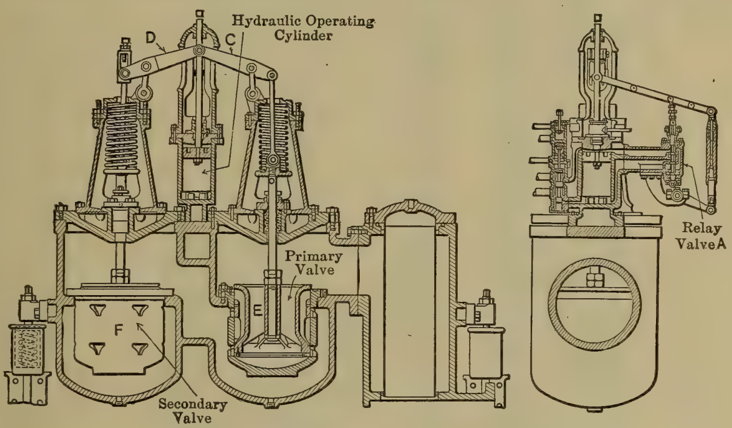


FIG. 275. Valve Gear with Oil Relay, Westinghouse-Parsons Turbine.

packing is alternately wire-drawn and checked so that the flow is greatly reduced.

227. Westinghouse Impulse-reaction or Double-flow Turbine. — In reaction turbines of the single-flow type, as illustrated in Fig. 269, the high-pressure portion dealing with the high-pressure incoming steam is the least efficient. This is due to the fact that the blade lengths are approximately proportional to the specific volume of the steam, and consequently the initial expansion in the turbine requires blade passages of very small dimensions. This results in greater leakage past the tips of the blades than in the low-pressure elements where the blades are long. Again, in the single-flow type the high-pressure balance piston occupies fully one half of the total balance-piston length of the shaft, while the low-pressure piston is $2\frac{1}{2}$ times the high-pressure diameter so that balance pistons occupy a large portion of the total bulk of the machine. By making the high-pressure element of the impulse type and by arranging the low-pressure reaction elements on either side as

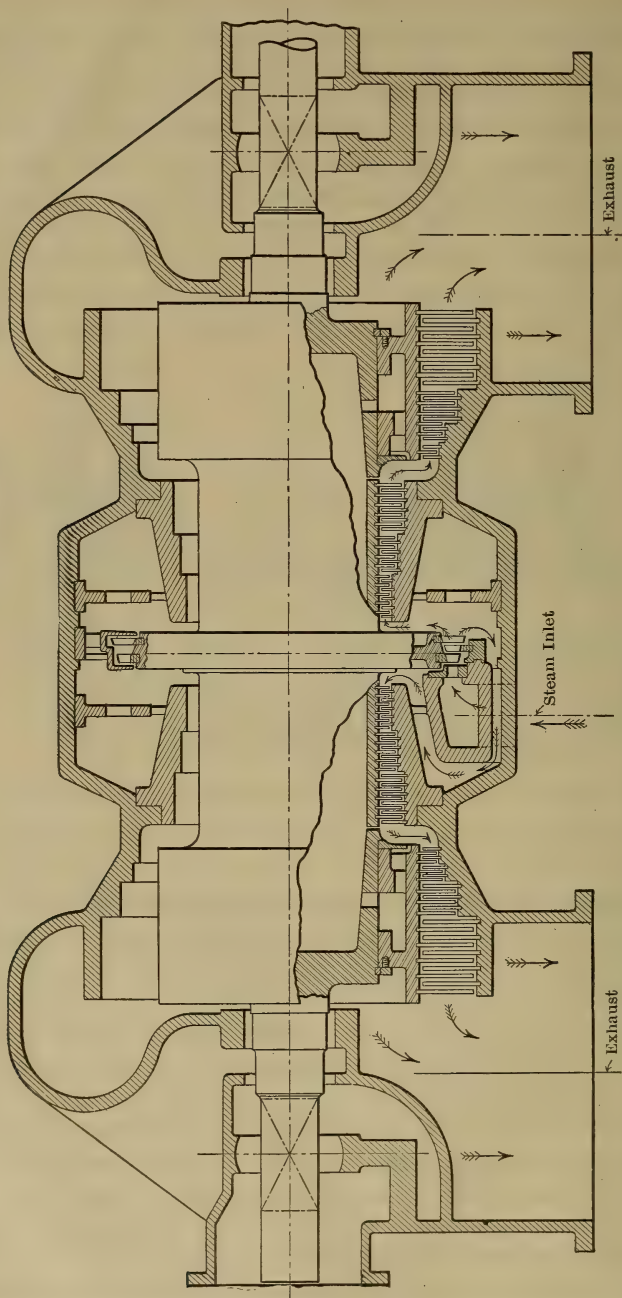


FIG. 276. Section through 10,000 Kilowatt Westinghouse Double-flow Turbine.

illustrated in Fig. 276 the efficiency may be increased and the bulk of the turbine may be greatly decreased. There are two rows of moving buckets upon the impulse wheel with an intermediate set of reversing blades, the operation being practically the same as in the first stage of a Curtis turbine. The drop in pressure in the nozzles is such that approximately 20 per cent of the total energy developed is absorbed by this impulse element. After leaving the impulse element the steam divides, one portion passing directly to the low-pressure blading at the left, while the rest passes through the hollow shell of the rotor to the similar pressure blades upon the right. As these sections are equal and symmetrical they counterbalance each other, and the balance or "dummy" pistons may be dispensed with. The advantages of the double-flow type over a single-flow unit of equal capacity are: (1) reduction of nearly 50 per cent in the shaft span between bearings; (2) the diameters of the casing and rotating part are more uniform, thus tending to greater rigidity; (3) a reduction of about 70 per cent in the bulk of the main parts of the machine, and (4) internal stresses due to high-pressure and high-temperature steam are avoided by isolating the incoming steam, without separate nozzle chambers.

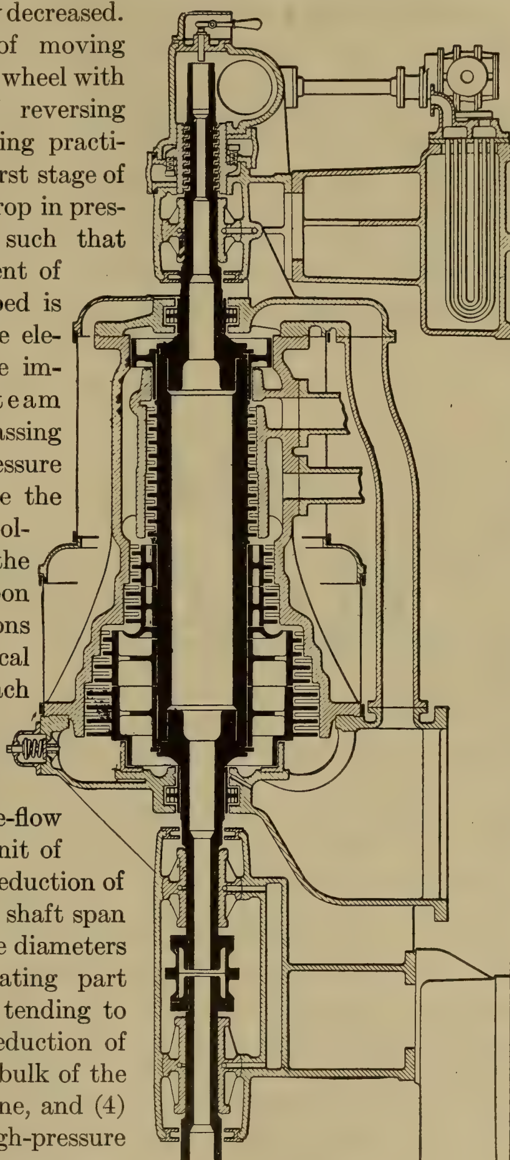


Fig. 277. Section through Allis-Chalmers Steam Turbine.

228. Allis-Chalmers Steam Turbine. — Fig. 277 shows a section through an Allis-Chalmers standard steam turbine, which is of the

Parsons type but differs from the original Parsons machine and the Westinghouse-Parsons construction principally in manufacturing details. In the older Parsons type, three balance pistons are placed at the high-pressure end. In the Allis-Chalmers design, the larger piston is placed at the low-pressure end of the rotor, behind the last row of blades, the other two remaining at the high-pressure end. This construction permits of a smaller balance piston and allows a smaller working clearance in the high-pressure and intermediate cylinders. In the Allis-Chalmers turbine the roots of the blades are dovetailed and fitted into a foundation ring, and the tips are incased in a channel-shaped shroud ring, thereby insuring a rigid and positively spaced construction. The governor is of the Parsons type, except that the main

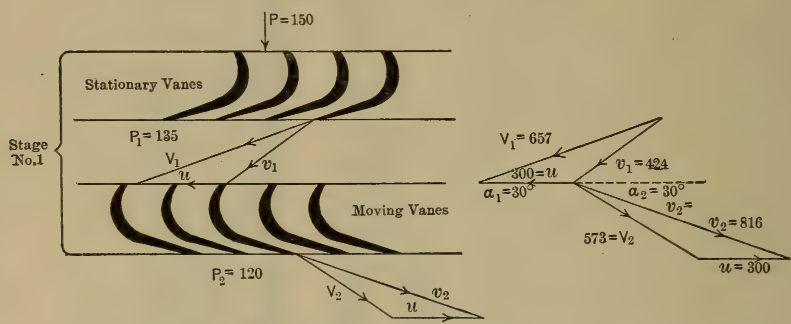


FIG. 278. Velocity Diagram. Westinghouse-Parsons Turbine.

valve and pilot valve are actuated by hydraulic instead of steam pressure. The bearings are of the self-adjusting ball-and-socket pattern and are kept "floating in oil" by a small pump geared to the turbine shaft. The oil is passed through a tubular cooler with water circulation after it leaves the bearings and is used over and over again.

229. Elementary Theory, Parsons Turbine. — Fig. 278 gives a diagrammatic arrangement of fixed and stationary blades in the first stages of a multi-stage ideal reaction turbine. The steam enters the stationary blades at practically zero velocity and is there partially expanded and impinges against the movable blades at velocity V_1, part of the energy of the steam being thus absorbed. In passing through the movable blades the steam is still further expanded and leaves at an absolute velocity V_2, exerting an additional pressure on the blades from the reaction. The steam enters the second set of stationary blades with velocity V_2 and is still further expanded to velocity V_3, and so on.

The energy imparted to the steam in the first set of stationary blades is

$$E_1 = \frac{W}{2g} \cdot V_1^2. \tag{157}$$

V_1 = absolute velocity of the steam leaving the blades.

The energy imparted to the steam in the first set of moving blades is

$$E_2 = \frac{W}{2g}(v_2^2 - v_1^2). \tag{158}$$

v_1 = relative velocity of the steam entering the moving blades.
v_2 = relative velocity of the steam leaving the moving blades.

The total energy acquired by the steam in the first stage is

$$E_1 + E_2.$$

The energy converted into work in this stage is

$$E = E_1 + E_2 - \frac{WV_2^2}{2g} \tag{159}$$

$$= (V_1^2 + v_2^2 - v_1^2 - V_2^2)\frac{W}{2g}. \tag{160}$$

V_2 = absolute velocity of the steam leaving the moving blades.

Each stage may be analyzed in a similar manner.

Example: A Westinghouse-Parsons turbine develops 1000 horse power on a steam consumption of 12 pounds of steam per horse-power hour. Initial steam pressure 150 pounds per square inch absolute; back pressure 1 pound per square inch absolute; drop in pressure in each set of fixed and moving blades 15 pounds per square inch; peripheral velocity 300 feet per second; $\alpha_1 = \alpha_2 = 30$ degrees. Compare the performance of the actual and ideal turbine.

Actual turbine:

Steam consumed per hour,

$$1000 \times 12 = 12{,}000 \text{ pounds.}$$

Steam consumed per second,

$$12{,}000 \div 3600 = 3.33 \text{ pounds.}$$

Horse power developed per pound of steam flowing per second,

$$1000 \div 3.33 = 300.$$

Kinetic energy per pound of steam,

$$300 \times 550 = 165{,}000 \text{ foot-pounds per second.}$$

Thermal efficiency,

$$E_t = \frac{2546}{12\,(1193.4 - 69.8)} = 18.8 \text{ per cent.}$$

Heat consumption, B.t.u. per horse-power hour per minute,

$$\frac{12\,(1193.4 - 69.8)}{60} = 224.7.$$

Efficiency ratio,

$$\frac{E_t}{E_r} = \frac{18.8}{28.1} = 66.9 \text{ per cent.}$$

Ideal turbine:

The velocity imparted to the steam in the first set of stationary blades due to the drop from 150 to 135 pounds per square inch is

$$\begin{aligned} V_1 &= 224\sqrt{H_1 - H_2} \\ &= 224\sqrt{1193.4 - 1184.8} \\ &= 657 \text{ feet per second.} \end{aligned}$$

Lay off the value of V_1 in direction and amount and combine with u, the peripheral velocity, Fig. 278. The resultant is v_1, the velocity of the steam relative to the blades. The angle between v_1 and the line of motion of the wheel will be the angle with the blade at entrance.

From the velocity diagram,

$$v_1 = 424.$$

E_2, the energy given up by one pound of steam in expanding from 135 to 120 pounds, is

$$\begin{aligned} E_2 &= 778\,(H_2 - H_3) \\ &= 778\,(1184.8 - 1175.1) \\ &= 7554 \text{ foot pounds per second.} \end{aligned}$$

Substitute $v_1 = 424$ and $E_2 = 7554$ in equation (158),

$$7554 = \frac{1}{64.4}\,(v_2^2 - 424^2),$$

$$v_2 = 816 \text{ feet per second.}$$

The resultant of v_2 and u is V_2, the absolute velocity of the steam leaving the moving blades of the first stage. From the diagram,

$$V_2 = 573 \text{ feet per second.}$$

The energy converted into work in the first stage is determined by substituting the proper values in equation (160), thus:

$$\begin{aligned} E &= (\overline{657}^2 + \overline{816}^2 - \overline{424}^2 - \overline{573}^2)\,\frac{1}{64.4} \\ &= 9150 \text{ foot-pounds per second.} \end{aligned}$$

The various stages may be analyzed in a similar manner.

The theoretical output of the entire turbine per pound of steam will be that corresponding to adiabatic expansion from a pressure of 150 to 1 pound absolute.

$$\begin{aligned} E &= 778\,(H_1 - H_n) \\ &= 778\,(1193.4 - 877.2) \\ &= 246{,}000 \text{ foot-pounds per second.} \end{aligned}$$

Horse power per pound of steam,

$$\text{H.P.} = \frac{246{,}000}{550} = 444.$$

Steam consumption per horse-power hour,

$$\frac{3600}{444} = 8.1 \text{ pounds.}$$

Thermal efficiency,

$$\begin{aligned} E_r &= \frac{1193.4 - 877.2}{1193.4 - 69.8} \\ &= 28.1 \text{ per cent.} \end{aligned}$$

The calculation of Parsons Turbines: Zeit. f. Turbine, May 10 and 20, 1912.

230. Low-pressure Mixed-pressure and Bleeder Turbines. — A promising field for the steam turbine is in its application as a secondary or low-pressure unit in connection with non-condensing or condensing engines, or, combined with a regenerator, in connection with engines using steam intermittently. Numerous examples may be cited showing great gains in both capacity and economy in existing power plants involving the abandonment of but a negligible part of the equipment and accomplishing this result with a minimum additional investment. The most notable installation, to date, of low-pressure turbines to condensing reciprocating engines is at the 59th Street Station of the Interborough Rapid Transit Co., New York. Three of the nine 7500-kw. Manhattan-type compound Corliss engines have been equipped with Curtis three-stage, low-pressure turbo-generators of equal capacity, and provision is made for the installation of six additional units. The low-pressure turbine is installed between the exhaust of the low-pressure cylinders and the condenser as shown in Fig. 279. Running with the engine the low-pressure turbine generator carries a variable load without governor regulation. The turbine generator takes care of the speed by automatically taking such a load as will keep the frequency in unison with that of the engine-driven generator. The turbine is equipped with the usual emergency speed-limit attachment for cutting off the steam supply should the speed exceed a predetermined limit. The performance of one set of engines, a high-pressure turbine of the equivalent total capacity, and that of the combined engine and low-pressure

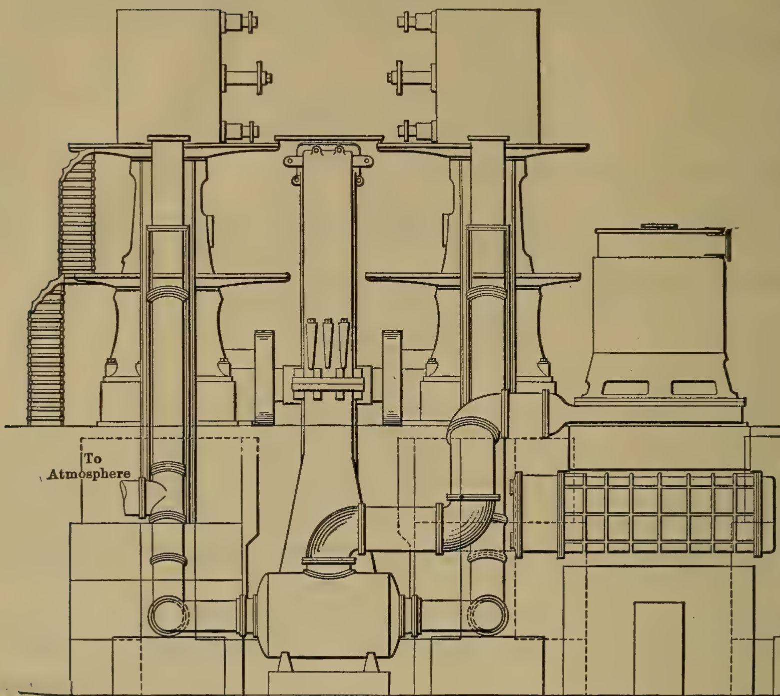


FIG. 279. Low-pressure Turbine Installation at the 59th Street Station of the Interborough Rapid Transit Company, New York.

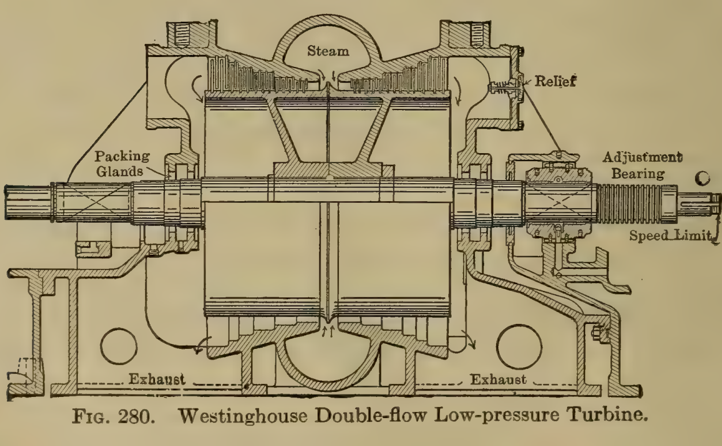


FIG. 280. Westinghouse Double-flow Low-pressure Turbine.

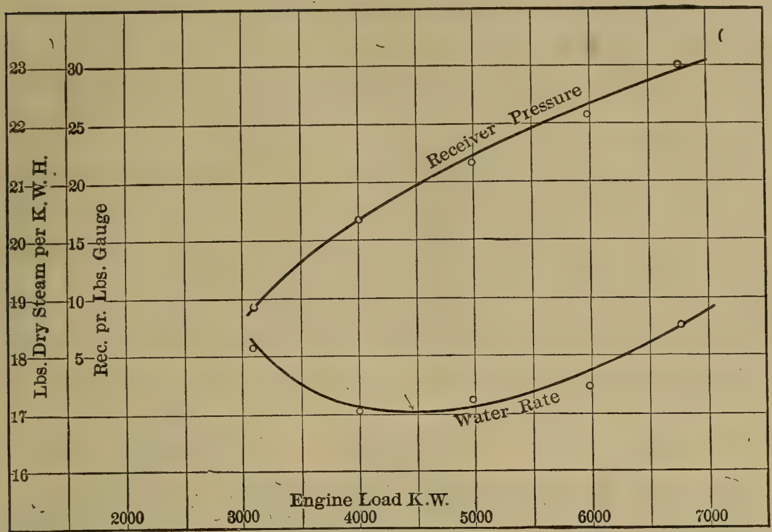


FIG. 281. Performance of 7500-Kw. Engine at 59th Street Station of Interborough Rapid Transit Company, New York, with Varying Receiver Pressure.

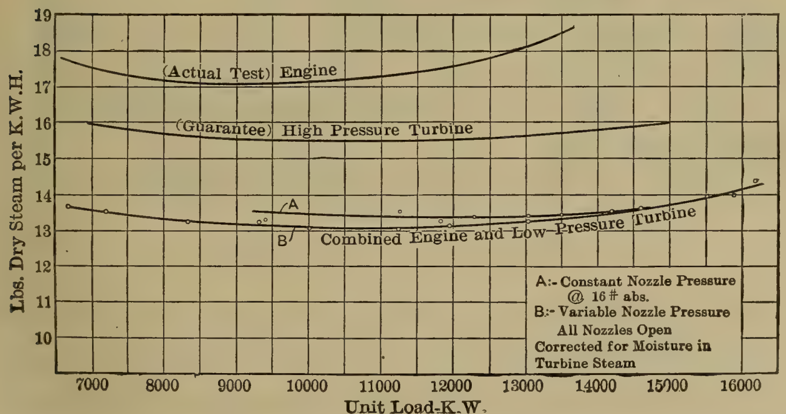


FIG. 282. Comparison of Economy Curves: 7500-Kw. High-pressure Turbine, 7500-Kw. Engine and Combined Engine and Low-pressure Turbine at the 59th Street Station of the Interborough Rapid Transit Company, New York.

turbine are illustrated in Fig. 282. The conclusions drawn from an exhaustive series of tests at this station are that the addition of low-pressure turbines effected:

a. An increase of 100 per cent in maximum capacity of plant.

b. An increase of 146 per cent in economic capacity of plant.

c. A saving of approximately 85 per cent of the condensed steam for return to the boiler.

d. An average improvement in economy of 13 per cent over the best high-pressure turbine results.

e. An average improvement in economy of 25 per cent (between the limits of 7000 kw. and 15,000 kw.) over the results obtained by the engine units alone.

f. An average unit thermal efficiency of 20.6 per cent between the limits of 6500 kw. and 15,500 kw.

If the turbine does not constantly require all the available exhaust from the high-pressure units the exhaust from the latter may be by-passed so that the high-pressure units may reap the benefit of the reduced back pressure when the turbine is carrying a light load. This bypass is controlled by the governing system. Similarly in case the exhaust from

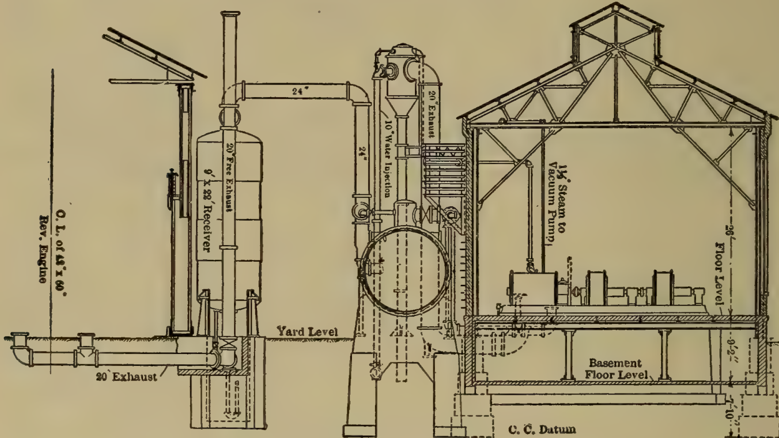


Fig. 283. Rateau Low-pressure Steam Turbine Installation.

the high-pressure units is in excess of the turbine requirements the bypass valve discharges part of the steam directly to the condenser.

Low-pressure turbines are frequently installed in connection with regenerative accumulators, to rolling-mill engines, steam hammers, and other appliances using steam intermittently, and have proved to be paying investments. A typical installation of this character is to be found at the South Chicago Division of the International Harvester Company. The front elevation of the turbine and regenerator installation is shown in Fig. 283 and the general arrangement of the regenerator is shown in Fig. 284. The regenerative accumulator is intended to regulate the intermittent flow of steam before it passes to the turbine. The steam collects and is condensed as it enters the apparatus and is again vaporized during the time when the exhaust of the engines diminishes or ceases. (See also Power & Engr., Nov. 8, 1910.)

The regenerator consists of a cylindrical boiler-steel shell divided into two similar chambers by a central horizontal diaphragm. In each compartment are a number of elliptical tubes A, each of which is perforated with a number of $\frac{3}{4}$-inch holes. The spaces surrounding the tubes and, under certain conditions, the tubes themselves are filled with water to a height of about four inches above the top of the upper tubes. Baffle plate B serves to separate the entrained moisture from the steam. The operation is as follows: Exhaust steam enters the apparatus at N, passes to the interior of the elliptical tubes, and escapes into the steam space through the perforations and thence to the turbine. When the supply of steam from the main engine ceases, the pressure in

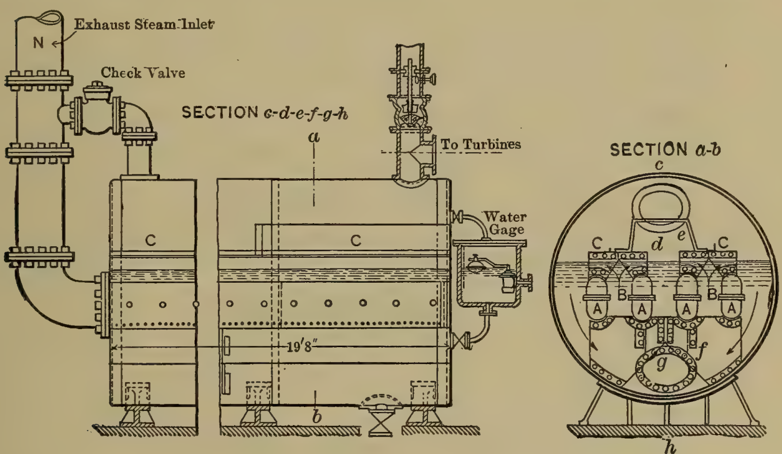


FIG. 284. Rateau Regenerator Accumulator.

the regenerator decreases, the water liberates part of the heat it has absorbed and a uniform flow of low-pressure steam is given off. The continued demand of the turbine reduces the pressure in the accumulator and causes the steam still retained in the tubes to escape, thereby maintaining the circulation of the water (indicated by arrowheads) and facilitating the liberation of steam. Suitable valves regulate the limits of pressure in the accumulator and prevent the return of water to the main engine.

In the size normally installed this type of accumulator will furnish a sufficient supply of steam for four minutes with exhaust entirely cut off. If the period is longer than four minutes it becomes necessary to admit live steam. Low-pressure turbines develop one electrical horse-power hour on a steam consumption of about 30 pounds with initial pressure of 15 pounds absolute and a back pressure of 1.5 pounds

absolute. Fig. 285 gives the performance of a typical Westinghouse low-pressure turbine for various vacua, initial pressure 15 pounds absolute.

The weight of water W required to operate the low-pressure turbine for a given period with a predetermined temperature drop may be calculated from the relationship

$$W = \frac{tsr}{q_1 - q_2}, \tag{161}$$

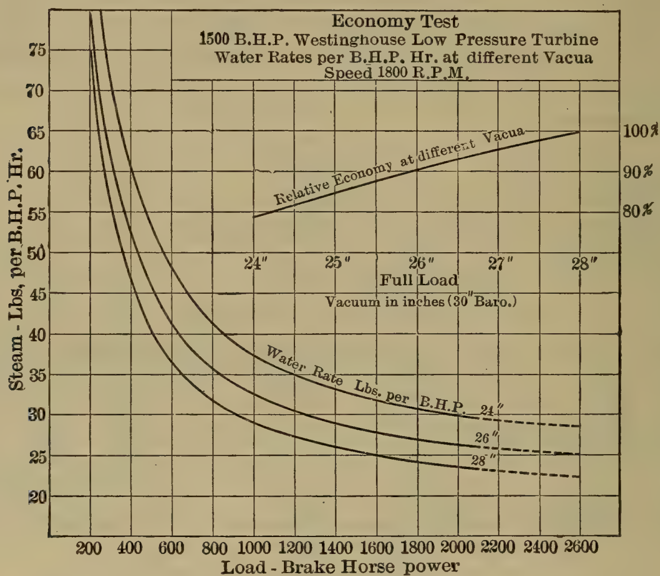


FIG. 285. Performance of Westinghouse Low-pressure Turbine.

in which

t = maximum number of minutes the exhaust supply may be entirely cut off;

s = water rate of the turbine, pounds per minute;

r = mean latent heat at regenerator pressure;

q_1 = heat of the liquid corresponding to maximum temperature of water in regenerator, degrees F.;

q_2 = heat of the liquid corresponding to minimum temperature of water in regenerator, degrees F.

If the regenerator is to absorb M pounds of exhaust steam in t minutes as in case of a sudden flux of exhaust the weight of water W_1 required is

$$W_1 = \frac{Mr}{q_1 - q_2}. \tag{162}$$

Example: Determine the weight of water to be stored in a regenerator to operate a 500-horse-power exhaust steam turbine for five minutes if the steam supply is entirely cut off; pressure drop 17 to 14 pounds absolute, turbine water rate 30 pounds per horse-power hour.

Here

$$t = 5, \quad s = \frac{500 \times 30}{60} = 250, \quad r = \frac{965.6 + 971.9}{2} = 968.8,$$

$$q_1 = 187.5, \quad q_2 = 177.5,$$

$$W = \frac{5 \times 250 \times 968.8}{187.5 - 177.5} = 121{,}100.$$

If the regenerator is to absorb 2000 pounds of the exhaust steam in five minutes during a period of sudden flux,

$$W_1 = \frac{2000 \times 968.8}{187.5 - 177.5} = 193{,}760.$$

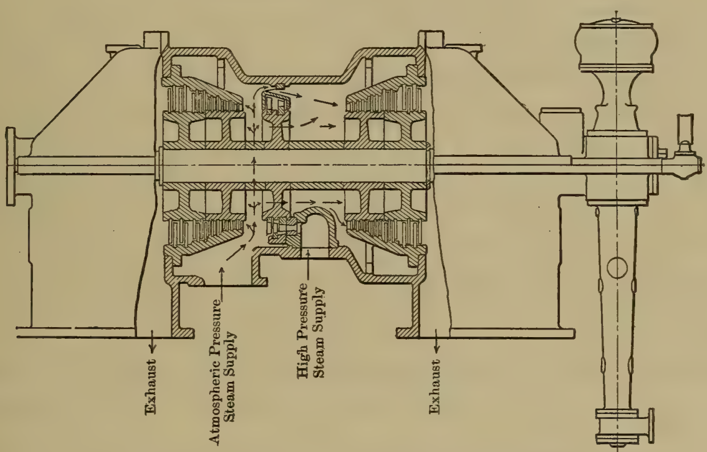


FIG. 286. Westinghouse Mixed-pressure Turbine.

Low-pressure turbines equipped with special expanding nozzles, or the equivalent, to receive steam at high pressure direct from the boilers are known as *mixed-pressure* turbines. With this construction the full power of the turbine can be developed with (1) all low-pressure steam, (2) all high-pressure steam, (3) any proportion of high- and low-pressure steam. In the Curtis mixed-pressure turbine this transition from all low pressure to all high pressure, through all the conditions intermediate between these extremes, is provided for automatically by the turbine governor; a deficiency of low-pressure steam causes the high-pressure nozzles to open automatically. With this arrangement it is not necessary for purposes of economy to proportion exactly the low-pressure

turbine to the amount of exhaust steam available, but within limits it may be made as large as the load demands.

Fig. 286 shows the general details of a Westinghouse mixed-pressure turbine.

Mixed-pressure turbines have been constructed in single units as large as 7000 kw. (Gen. Elec. Rev., July, 1912.)

In compound condensing engine plants it has been general practice to draw steam from the receiver for heating and manufacturing purposes. This same result is effected in the *bleeder type* of turbine, through the agency of a pressure-controlled valve placed between the high- and low-pressure section of the turbine and which automatically diverts the required amount of steam at a predetermined pressure to the heating system.

Mixed Pressure Turbines and Engines: Power, Feb. 1, 1910; Eng. Rec., Feb. 19, 1910; Eng. Mag., Apr., 1909; National Engr., May, 1912.

Geared Turbines: Engng., May 24, 1912.

Heating in Connection with Steam Turbines: Power, Sept. 17, 1912, p. 426.

231. Advantages of the Steam Turbine. — The principal advantages of the steam turbine are: (1) simplicity; (2) economy of space and foundation; (3) absence of oil in condensed steam; (4) freedom from vibration; (5) uniform angular velocity; and (6) high efficiencies for large variations in load. The reciprocating engine is well adapted for pumping stations, direct-current generators, compressor plants, hoisting engines, and the like, requiring low angular velocity, but its place is being rapidly taken by the steam turbine for alternating-current dynamos, centrifugal pumps and blowers requiring high angular velocity. The recent development of high-efficiency speed-reduction gearing makes it possible for the turbines to compete with the engines for low-speed work.

Simplicity. — Although composed of a large number of parts as compared with a reciprocating engine of the same capacity, there are few moving parts and rubbing surfaces. The only contact between rotor and stator is in the main bearings, and the problem of lubrication is therefore a simple one. The absence of pistons, stuffing boxes, dash pots, etc., reduces the cost of maintenance and attendance to a minimum and limits the possibility of leakage.

Economy of Space and Foundation. — The floor space required by practically all types of turbines is considerably less than the space requirements of piston engines. Vertical three-cylinder compound Corliss engines of the New York Edison type require the least floor space of any large slow-speed reciprocating engines, but take up about twice the space of a Parsons turbine installation of the same size. With

high-speed engines of the Willans central-valve type the comparative economy in space is less marked.

The weight of the steam turbine is very small compared with a reciprocating engine of the same horse power. The New York Edison engine and generators weigh more than eight times as much as a turbine installation of equal capacity. The turbine, for this reason, and also because of the total absence of vibration, requires a relatively light foundation. In many instances the foundation consists of steel beams with concrete arches sprung between them resting upon the floor, and the basement underneath may be used for the condenser instead of the massive foundation required for the reciprocating engine. Engines are seldom constructed in sizes above 10,000 horse power, whereas single turbine units of 20,000 kw. are not uncommon and a Parsons turbine of 25,000 kw. normal capacity is now being installed in the Fisk Street Station of the Commonwealth Edison Co., Chicago.

Absence of Oil in Condensed Steam. — As the steam turbine requires no internal lubrication, oil does not come in contact with the steam, and the condensed steam from the surface condensers is available for boiler-feeding purposes without purification. In many cases the reuse of condensed steam effects a large saving in cost of feed water and in expense for maintenance and cleaning of boilers. The amount of entrained air is reduced to a minimum and consequently the work of air pumps is lessened.

Regulation. — The variable pressure at the crank pin of a reciprocating engine necessitates the use of a heavy flywheel to keep the instantaneous angular fluctuation within practical limits. In the steam turbine the motion is purely rotary and a flywheel is not necessary. In the former there are always instantaneous variations in velocity during each revolution, even with constant load, while in the latter the speed is practically constant. A number of published tests of Parsons and Curtis turbines show an average fluctuation of 2 per cent from no load to full load and 3 per cent from no load to 100-per-cent overload. Although closer regulation than this is possible, it is not deemed necessary, particularly in alternating-current work where a comparatively wide range is desirable for parallel operation.

Overload Capacity. — The overload capacity of any prime mover depends entirely upon the designation of the rated load. The maximum economy of the average piston engine lies between 0.7 and full load, and for this reason the *rated* load refers usually to this maximum economical load. Evidently if the engine is rated under its maximum possible output it is capable of *overload.* Under the existing system of rating the average piston engine is capable of operating with overloads of

25 to 50 per cent. According to the old rating the steam turbine was capable of overloads ranging from 100 to 200 per cent and much confusion arose in determining the station load factor. Current turbine practice gives as the normal rating the maximum continuous load which can be carried for 24 hours when under control of the primary valves. Through the agency of the secondary valves overloads of 50 per cent or more are possible. The steam economy of the turbine is superior to that of the engine for overloads.

232. Efficiency and Economy of Steam Turbines. — As far as steam consumption is concerned there is practically no difference between the performance of standard high-grade piston engines and that of first-class steam turbines (both using saturated steam) for sizes under 2000 kilowatts, the choice depending more upon variable load characteristics and space requirements than upon heat economy. Engines of the uniflow type are more economical in steam consumption than turbines of equivalent capacity and piston engines using highly superheated steam are decidedly more economical of fuel than turbines under the best conditions of operation, but heat economy is only one of the items entering into the ultimate cost of power. In a general sense the turbine gives a flatter load characteristic with saturated steam than the standard piston engine * and for this reason is better adapted to variable loads, but this advantage disappears with the use of highly superheated steam. For sizes over 2000 kilowatts the turbine is in a class of its own and piston engines above this size are seldom found in modern stationary practice. A comparison of Fig. 198 showing typical economy curves of high-speed single-valve non-condensing engines, and of Fig. 289 showing similar curves for small non-condensing turbines is somewhat in favor of the piston engine, though the difference is small; whereas a comparison of the turbine and engine curves in Fig. 282, showing the performance of very large units, is decidedly in favor of the turbine. Any number of individual tests may be cited showing superiority in fuel consumption of the piston engine over that of a turbine of equivalent capacity and vice versa, but when the machines are designed for the same operating conditions the results are practically the same for all sizes under 2000 kilowatts. Tables 58 to 74 give the general condition of operation and the steam consumption of exceptionally good piston engines of various sizes and types, and Table 75 similar data of first-class turbines. A study of these tables will show that the choice must be based on other factors than the steam consumption. In a general sense, the piston engine is superior to the turbine for high back pressures, slow rotative speeds and heavy starting torques, while the turbine has practically superseded the engine for

* This applies to very large units only.

TABLE 75.

EXAMPLES OF STEAM TURBINE PERFORMANCES.

Index.	Make of Turbine.	Reference.	Test Load, Kilowatts.	Rated Capacity, Kilowatts.	R.P.M.	Initial Pressure, Lbs. per Sq. In. Absolute.	Back Pressure, Inches of Mercury Absolute.	Degree of Superheat Fahr.	Water Rate, Lbs. per Kilowatt Hour.	B.t.u. per Electrical Horse Power per Minute.	Efficiency Ratio, Per Cent.
1	Allis-Chalmers	Sibley Journal, Jan., 1911	4,300	4,000	1800	186.4	1.96	10.85	14.02	203.1	68.4
2	Do	Power, 1/12/12	3,850	3,750	1800	164.7	2.01	125.0	15.40	228.8	63.5
3	Curtis	Jour. A.S.M.E., Aug., 1912	12,460	12,000	720	207.5	2.48	191.0	13.63	206.0	68.2
4	Do	....do	12,000	12,000	750	200.0	2.00	125.0	14.22	211.0	66.2
5	Do	Trans. A.S.M.E., Vol. 32	8,880	8,000	....	192.5	1.72	108.5	15.05	223.1	63.1
6	Do	....do	8,775	9,000	750	194.0	1.89	72.9	15.95	233.0	61.0
7	Do	Journ. A.S.M.E., Aug., 1912	7,526	7,500	720	184.0	1.35	134.0	13.73	204.0	66.6
8	Do	....	2,023	2,000	900	181.0	2.24	207.0	15.02	231.0	61.6
9	Do	G. H. Barrus	519	500	1850	150.0	0.92	289.6	15.91	259.0	51.0
10	Do	Street Ry. Jour., Sept., 04	100	100	3600	140.0	2.44	0.0	25.70	353.0	60.5
11	Curtis-Parsons	Journ. A.S.M.E., May, 1912	2,128	2,000	1500	156.2	2.03	120.4	13.82	204.6	71.8
12	Curtis-Rateau	....do	6,518	....	1220	198.7	0.64	219.7	11.43	182.0	68.7
13	De Laval	Eng. Rec., 8/2/02	352	300	750	213.0	2.85	84.0	13.94*	272.0*	53.0
14	Do	Am. Elect., Aug., 1905	160	150	1200	161.0	3.56	0.0	26.49	324.0*	44.0
15	Do	Trans. A.S.M.E., Vol. 25	30	30	....	140.0	5.50	0.0	22.50*	406.0*	43.5
16	Kerr	Shop tests	500	500	3600	165.0	2.00	100.0	13.00*	257.0*	57.0
17	Do	....do	500	500	3600	165.0	30.00	100.0	25.00*	450.0*	54.5
18	Do	....do	250	250	3600	165.0	2.00	100.0	14.00*	276.0*	53.0
19	Do	....do	250	250	3600	165.0	30.00	100.0	25.50*	455.0*	53.4
20	Do	....do	100	100	3600	165.0	30.00	100.0	28.50*	508.0*	47.9
21	Ljingstrom	Engng., 4/19/12	972	1,000	3000	176.7	1.34	285.4	11.55	186.0	71.3
22	Westinghouse Double Flow	Journ. A.S.M.E., May, 1912	11,466	10,000	750	191.7	1.85	105.6	14.45	213.9	65.5
23	Do	....do	9,173	9,000	1800	181.7	2.11	59.0	14.57	200.4	68.9
24	Do	Elec. Rev., 6/23/11	5,066	5,000	1500	190.2	1.24	173.5	13.00	200.0	67.0
25	Westinghouse-Parsons Std.	Shop tests	3,638	3,500	1500	180.5	2.40	60.0	15.90	228.0	65.4
26	Do	....do	2,257	2,000	3600	190.0	4.00	Wet	13.90*	252.0*	62.9
27	Do	....do	1,012	1,000	3583	165.0	6.30	105.0	19.00*	362.0*	55.3
28	Do	....do	821	750	3578	187.8	44.00	0.3	24.90*	414.0*	63.8

* B.h.p.

large central station units and for auxiliaries requiring high rotative speed. Recent tests of the Melville reduction gear (Machinery, Feb., 1910) show exceptionally high efficiencies for sizes as large as 6000

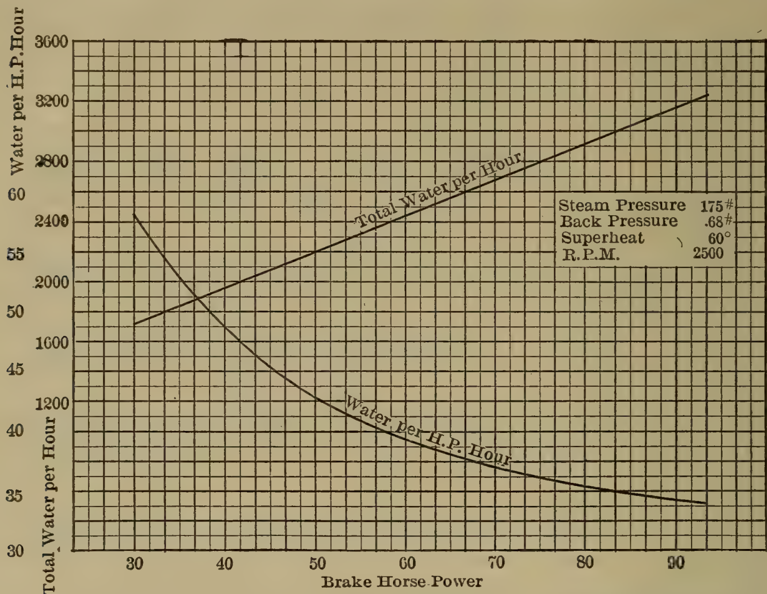


Fig. 287. Typical Performance of a 90-horse-power Terry Steam Turbine.

kilowatts, and it is not unlikely that the turbine equipped with this device will offset the low rotative speed factor of the piston engine.

If the tests of steam turbines and piston engines could be made at some standard initial pressure, back pressure and quality or superheat,

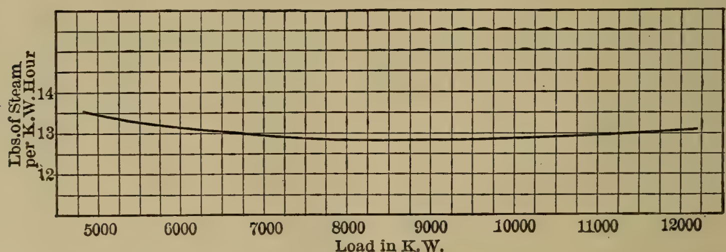


Fig. 288. Typical Performance of 9000-kilowatt Curtis Turbine; 200 pounds Gauge Pressure, 125 degrees Superheat, 29 Inches Vacuum.

then a comparison could readily be made, but both types of prime movers are designed to give the best results for special operating conditions, and any marked departure from these conditions will result in loss of economy. It is frequently desired, however, to make a

comparison between the economy of the different machines, and the following methods are in vogue:

(1) Steam consumption under assumed conditions.

(2) Heat consumption per unit output per minute above the ideal feed-water temperature.

(3) Efficiency ratio or ratio of actual to ideal.

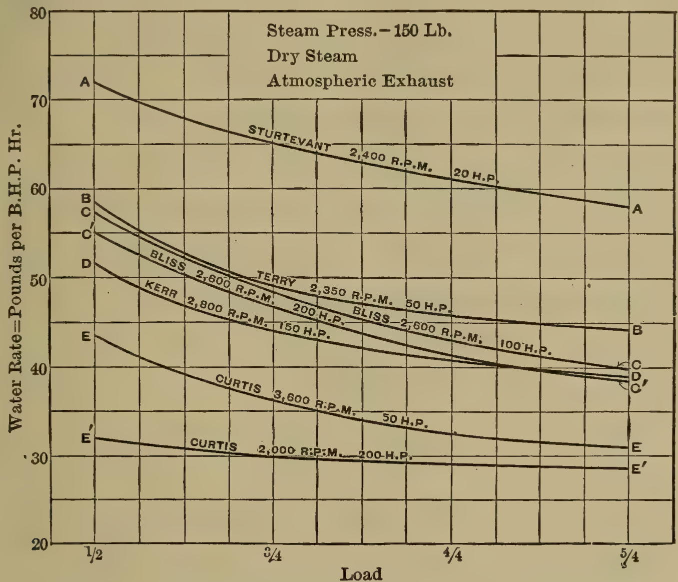


FIG. 289. Economy Tests of Small Non-condensing Turbines.

Standard Correction Curves:

This method for comparing engines or turbines or both is best illustrated by a specific example: Suppose it is required to compare the full-load performance of a 125-kilowatt direct-connected piston engine with that of a 125-kilowatt turbo-generator with operating conditions as follows:

	Steam Consumption, Lbs. per Kw.-Hour.	Initial Pressure, Lbs. Absolute.	Vacuum, Inches of Hg.	Superheat, Deg. F.
Engine..................	25.0	160	25.5	0
Turbine.................	22.7	110	28.0	125

Manufacturers of steam turbines have provided correction curves as illustrated in Fig. 290, showing the influence of varying vacuum, superheat and pressures on the steam consumption.* From curve *B*, we

* These curves are drawn to a much larger scale than the reproduction given here.

find that the steam consumption of the turbine should be decreased 2.5 pounds to give the equivalent at 160 pounds initial pressure; from curve A it should be increased 2.5 pounds to give the equivalent at 25.5 inches of vacuum, and from curve C it should be increased 2.5 pounds to give the equivalent at 0 degree superheat. The full-load steam consumption for the turbine under the engine conditions is therefore $22.7 - 2.5 + 2.5 + 2.5 = 25.2$ pounds per kilowatt-hour.

The *ratio* method is also used in this connection, thus: The full-load steam consumption at 160 pounds pressure, curve B, Fig. 290, is multiplied by the ratio $\frac{25}{27.5}$ to give the equivalent consumption at 110 pounds (25 is the steam consumption at 160 pounds and 27.5 the consumption at 110 pounds). Similarly the correction ratio to change the consumption at 28 inches of vacuum to 25.5 is $\frac{25.5}{25}$, and to correct 125 degrees F. superheat to 0 degree F. is $\frac{25}{22.5}$.

SUMMARY.

$$\text{Pressure correction } \frac{25}{27.5} = 0.91 = -\ 9 \text{ per cent.}$$

$$\text{Vacuum correction } \frac{27.5}{25} = 1.10 = 10 \text{ per cent.}$$

$$\text{Superheat correction } \frac{25}{22.5} = 1.11 = 11 \text{ per cent.}$$

Net correction 12 per cent.

Corrected steam consumption $= 22.7 + 0.12 \times 22.7 = 25.4$ pounds per kilowatt-hour.

The ratio method is generally used if the difference between the corrected steam consumption and that of the correction curves for the same conditions is greater than 5 per cent ("The Steam Turbine," Moyer, p. 128).

This ratio method for correcting steam consumption at full load may be used without appreciable error for half to one and one half load and is the only practical method for quarter load. (Engineering, London, March 2, 1906.)

Heat Consumption:

The heat consumption B.t.u. per unit output per minute above the ideal feed-water temperature may be expressed

$$\frac{W\,(H_1 - q_2)}{60}. \quad \text{See equation (136).}$$

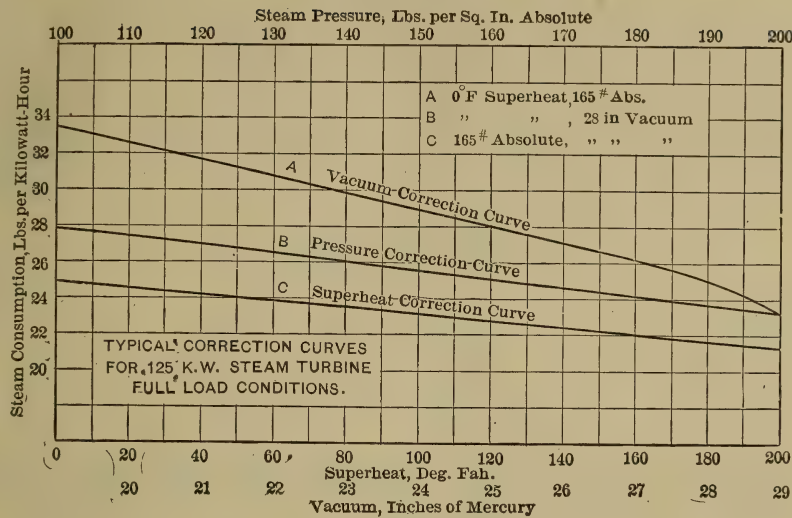


Fig. 290.

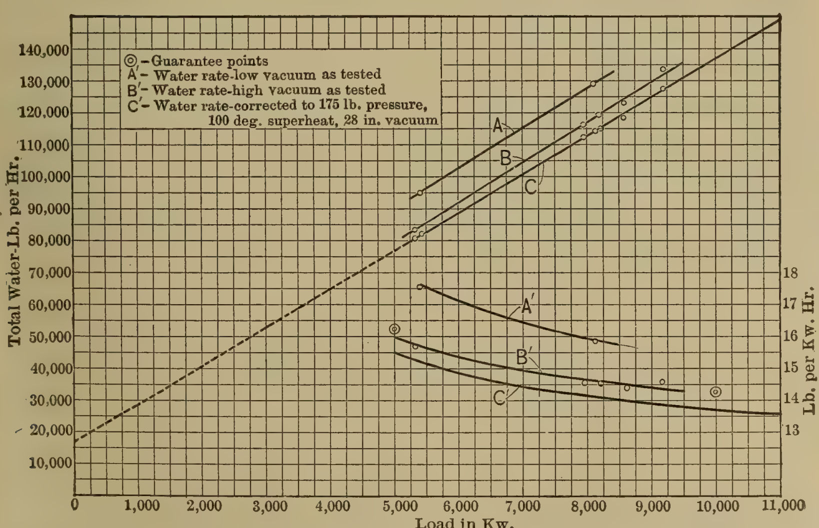


Fig. 291. Performance of 10,000-Kilowatt Westinghouse Double-flow Turbine, City Electric Co., San Francisco, Cal.

For the case cited above

Engine, $\frac{25\,(1194.1 - 98)}{60} = 455$ B.t.u.

Turbine, $\frac{22.7\,(1264.2 - 70)}{60} = 451$ B.t.u.

Efficiency Ratio:

The efficiency ratio, or the extent to which the theoretical possibilities are realized, may be expressed

$$E_r = \frac{2546 \times 1.34}{W\,(H_1 - H_2)}.$$ See equation (139).

For the case cited above

Engine, $\frac{2546 \times 1.34}{25\,(1194.1 - 915)} = 0.49.$

Turbine, $\frac{2546 \times 1.34}{22.7\,(1264.2 - 915.3)} = 0.43.$

In the assumed case the turbine is the more economical in heat consumption, but the engine is the more perfect of the two as far as theoretical possibilities are concerned.

A Recent Comparison of Turbines and Engines: Eng. Rec., Feb. 12, 1911; Power, Feb. 1, 1910.

Comparing Steam Turbine Tests: Power, May 2, 1911.

Testing Steam Turbo-Generators: Proc. A.I.E.E., Dec., 1910.

Test of a 10,000-*Kilowatt Turbo-Generator Set:* Jour. A.S.M.E., Dec., 1910.

The Present State of Development of Large Steam Turbines: Jour. A.S.M.E., May, 1912.

233. First Cost. — Steam turbines, generally speaking, are about 10 per cent lower in first cost than high-grade compound engines of equivalent power though the price depends largely upon the type and conditions for which they are designed. The figures in Table 76 refer to the cost of the high-pressure turbines direct connected to generators exclusive of auxiliaries and give some idea of the unit cost for various sizes.

TABLE 76.

APPROXIMATE COST OF STEAM TURBINES AND GENERATORS.

In Dollars per Kilowatt. Rated Capacity.

	Kilowatts.										
	25	50	75	100	200	300	500	1000	2000	4000	6000
Direct current:											
Non-condensing	55	47	42	38	32	32	....	....	......	......	......
Condensing	60	51	46	43	36	36	....	....	......	......	......
Alternating current:											
25 cycles	....	....	....	....	....	35	32	28	25	21	20
60 cycles	....	....	....	....	....	35	30	27	25	21	20

234. Cost of Operation.— Data pertaining to first cost of operating steam-turbine and reciprocating-engine plants and combinations of both will be found in Chapter XVIII. The following table, contributed by H. G. Stott, Superintendent Motive Power of the Interborough Rapid Transit Company, New York, gives an excellent comparison of the *relative* maintenance and operating costs (to date) of the three types of steam power plants as applied to large central stations for electric street railways.

TABLE 77.

RELATIVE COSTS PER KILOWATT-HOUR. DISTRIBUTION OF MAINTENANCE AND OPERATION.

	Reciprocating Steam Plant.	Steam Turbine Plant.	Reciprocating Engines and Low-pressure Steam Turbines.
Maintenance.			
1. Engine room, mechanical	2.59	0.51	1.55
2. Boiler or producer room	4.65	4.33	3.55
3. Coal and ash handling apparatus	0.58	0.54	0.44
4. Electrical apparatus	1.13	1.13	1.13
Operation.			
5. Coal	61.70	55.53	46.48
6. Water	7.20	0.65	0.61
7. Engine room labor	6.75	1.36	4.06
8. Boiler or producer room labor	7.20	6.74	5.50
9. Coal and ash handling labor	2.28	2.13	1.75
10. Ash removal	1.07	0.95	0.81
11. Electrical labor	2.54	2.54	2.54
12. Engine room lubrication	1.78	0.35	1.02
13. Engine room waste, etc.	0.30	0.30	0.30
14. Boiler room lubrication, etc.	0.17	0.17	0.17
Relative operating cost, per cent	100.00	77.23	69.91
Relative investment, per cent	100.00	75.00	80.00
Probable average cost per kw	125.00	93.75	100.00
Probable fixed charges	11%	11%	11%

For steam-turbine plants larger than 60,000 kilowatt the cost per kilowatt may be reduced to $65.00.

235. Influence of Superheat. — The use of superheated steam increases the economy of the reciprocating engine about 1 per cent for every 10 to 20 degrees of superheat, depending upon the conditions of operation, the gain being due mainly to the reduction of cylinder condensation. Cylinder condensation is reduced not only because of the excess heat available for the evaporation of moisture but also because superheated steam has a lower conductivity than wet steam, and less heat is given up to the cylinder walls for the same difference of temperature. In the steam turbine this difference of temperature is much smaller, since high- and low-pressure steam do not alternately come in

contact with the same surface as is the case with the reciprocating engine, and the time of contact is considerably less, due to the comparatively high velocities. With a well-lagged casing, therefore, the condensation due to this cause is insignificant compared with that of the reciprocating engine, and the beneficial effect of superheat is much more pronounced. Friction of the steam, which in the reciprocating engine is negligible, and which may be a source of considerable loss in the turbine, is greatly reduced by the use of superheated steam, as is also the "windage" loss due to the rapid revolution of the wheels.

The problem of cylinder lubrication is sometimes a difficult one in steam engines using a high degree of superheat, and trouble is frequently experienced due to the unequal expansion of the metal. In the steam turbine the latter difficulty is not so pronounced and no internal lubrication is necessary, hence a high degree of superheat is permissible. For maximum economy the steam at the end of expansion should be free from moisture. Assuming purely adiabatic expansion, the steam in expanding from 165 pounds to 1 pound absolute would have to be superheated about 1300 degrees F., giving the steam an actual temperature of 1800 degrees F. A study of some 100 tests made in this country gives about 250 degrees superheat as a maximum and 100 degrees to 150 degrees F. as an average. In Europe reciprocating engines are operating with superheat as high as 450 degrees F. and turbines 300 degrees F.* The additional fixed and operating costs of superheating must be considered in determining the net gain, since the decrease in steam consumption does not represent the actual saving. With pressures of 175 pounds gauge or less, and not to exceed 200 degrees F. superheat, the net gain has in most cases proved a substantial one. With higher temperatures and pressures the cost of maintaining the superheat may increase more rapidly than the saving in steam consumption, until a limit is reached beyond which no advantage is gained. This is illustrated in Fig. 292. (From paper read by E. D. Dreyfus before the Railway Club of Pittsburgh, May 20, 1910.)

236. Influence of High Vacua. — The possible economy of the reciprocating engine is greatly restricted by its limited range of expansion. Cylinders cannot be profitably designed to accommodate the rapid increase in the volume of steam when expanded to very low pressures. For example, the specific volume of 1 pound of steam under a vacuum of 29 inches (referred to a 30-inch barometer) is about 667 cubic feet, or nearly double its volume under a vacuum of 28 inches. Usually the exhaust is opened at a pressure of 6 or 8 pounds absolute and con-

* For results of European turbine tests with various degrees of superheat see Power & Engr., Feb. 14, 1911, p. 288.

sequently a large proportion of the available energy is lost. The lower vacuum in the exhaust pipe, therefore, serves only to diminish the back pressure and does not affect the completeness of expansion. Even if it were practical to expand to 1 pound absolute, the increased condensation in the reciprocating engine would offset any gain due to expansion unless the steam were highly superheated. A study of a number of tests of reciprocating engines shows a slight improvement due to increasing the vacuum beyond 26 inches. Tests of steam turbines show

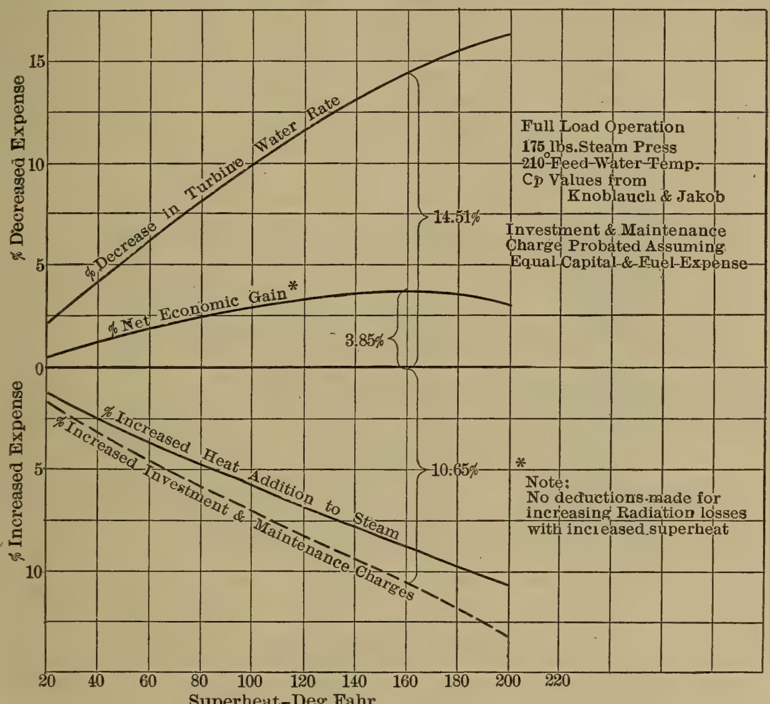


FIG. 292. Influence of Superheat on Overall Economy of Operation.

a decrease of 3 to 4 per cent in steam consumption for each inch increase of vacuum between 25 and 29 inches, for with a well-lagged casing cylinder condensation is practically absent, since the high- and low-temperature steam do not alternately come in contact with the metallic surfaces as is the case with the reciprocating engine.

Since the volume of the steam increases very rapidly with the decrease in back pressure the corresponding capacity and power required by the air and circulating pumps becomes proportionately larger. There is consequently a point where the improvement in steam economy fails to exceed the increased power demanded by the auxiliaries. This is

illustrated in Fig. 293 taken from the discussion by E. D. Dreyfus on a paper entitled "Test of a 10,000-kw. Steam Turbine," S. L. Naphtaly, Proc. A.S.M.E., Dec., 1910. The following notes refer to Fig. 293:

200-kilowatt turbine using surface condensers supplied with average injection water at 55 degrees F. Steam conditions 175 pounds, 100 degrees superheat. Most economic arrangement of auxiliaries selected for each vacuum to give best heat balance. Relation between investment and fuel economy pro-rated on the assumption of the fixed charges being one half of fuel expense burning coal at $3.00 per ton.

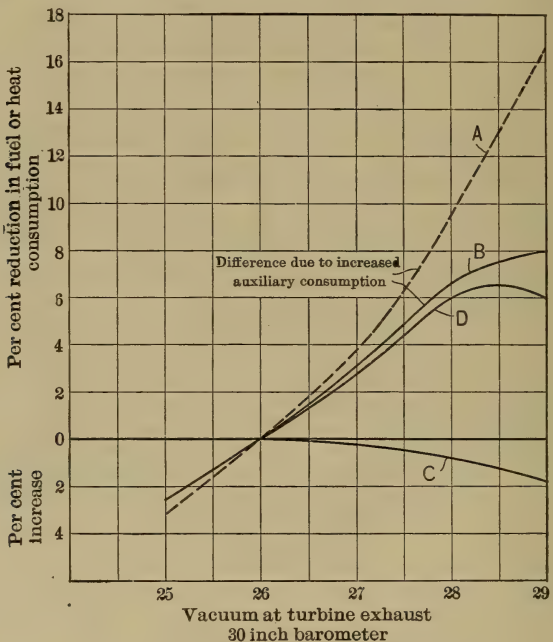


FIG. 293. Influence of Vacuum on Power.

A = Actual reduction at the turbine.
B = Net reduction in plant fuel consumption.
C = Equated cost (greater fixed charges and maintenance) of obtaining the higher vacuum.
D = Final plant improvement.

237. Tesla Bladeless Turbine. — Fig. 294 shows a section through a 200-horse-power experimental turbine designed by Nikola Tesla for which extravagant claims have been made. It consists of a rotor composed of 25 steel disks (each $\frac{1}{32}$ inch thick and arranged on the shaft so that the length of the shaft covered by the disks is approximately 3.5 inches) revolving in a plain cylindrical casing. There are no guide

plates or vanes and the viscosity and adhesion of the steam is depended upon for driving the rotor instead of impulse and reaction as in the standard type of turbine. Steam flows from the circumference to the center, and, when the rotor is at rest, flows by a short curved path, as indicated by the line in the end view, across the face of the disk. When the rotor is up to speed the steam passes to the exhaust in a spiral path from 12 to 16 feet in length. Since the direction of rotation is determined solely by the direction of the entering jet it is only necessary to change the direction of the latter to effect complete reversal of the rotor. Mr. Tesla states that a 200-horse-power turbine of this type has attained a performance of 38 pounds per horse-power hour, initial pressure 125 pounds gauge, atmospheric exhaust, 9000 r.p.m. (Prac. Engineer, U. S., Dec., 1911, p. 852.) The space occupied by this unit is only

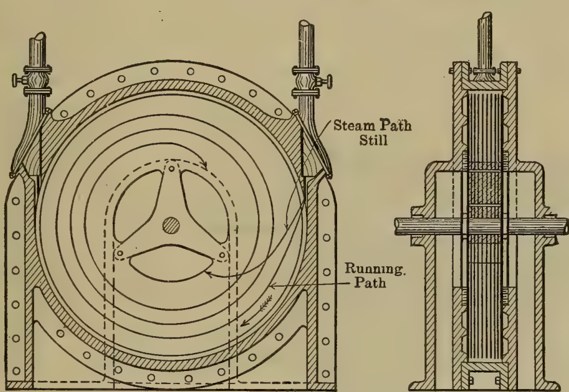


Fig. 294. Tesla Bladeless Turbine.

2 feet by 3 feet and 2 feet high and the weight of the engine alone is 2 pounds per horse power developed. If the development of the future bears out Mr. Tesla's prediction this type of prime mover will revolutionize the steam-turbine industry. At this writing (Nov. 1912), however, very little information is available concerning the present status of this apparatus.

238. "Spiro" Turbine. — Fig. 295 gives the general details of a new type of steam motor which is a sort of compromise between the rotary engine and the steam turbine. It consists essentially of a pair of herringbone gears revolving in a twin cylindrical casing. Steam enters space *a*, Fig. 295, through ports *pp* and presses upon the gear teeth, driving them forward. The volume is increased from that indicated at *a* to that shown at *b*, *c*, *d*, *e*, and *f* and the energy produced is the product of the pressure and volume. Exhaust occurs when the ends of the grooves in which the action lies pass the line of contact so that

they are no longer closed by the teeth of the opposite gear. Rotation is effected by both pressure and impulse, although no attempt is made to produce a considerable pressure drop between the steam chest and the admission pressure. The load may be varied by throttling or by cutting off the steam supply. The "Spiro" is built in various sizes ranging from 1 to 200 horse power and operates at 2000 to 3000 r.p.m. The following tests give an idea of the economy effected by this type of motor. (Power, Feb. 6, 1912, p. 188.)

	Test 1.	Test 2.
Boiler pressure, pounds gauge	120	130
Inlet pressure, pounds gauge	101.5	115
Back pressure, pounds gauge	Atmos.	Atmos.
Horse power developed	25.3	151
R.p.m.	2450	2710
Steam, pounds per horse-power hour	53.2	31.8

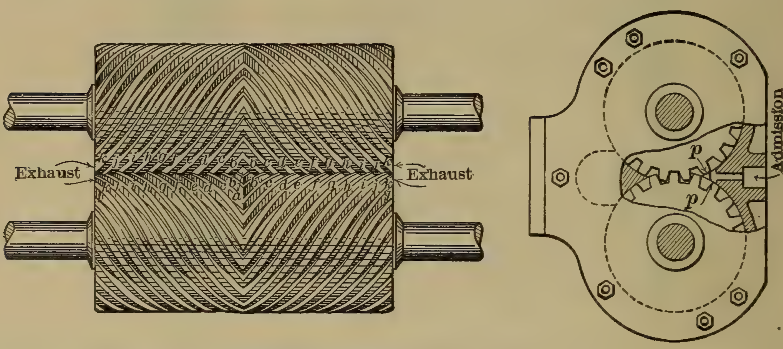


FIG. 295. The "Spiro" Turbine.

CHAPTER XI.

CONDENSERS.

239. General. — A pound of dry steam at atmospheric pressure (29.92 inches mercury) occupies a volume of 26.79 cubic feet. Suppose these 26.79 cubic feet of steam were contained in a closed vessel, and that the steam was subsequently condensed and its temperature lowered by suitable means to, say, 110 degrees F. The condensed steam would occupy only about $\frac{1}{1700}$ of its original volume, and the pressure would fall to 2.59 inches of mercury, the latter pressure being due to the tension of the aqueous vapor at the given temperature. That is to say, the best vacuum theoretically attainable under the given conditions would be $29.92 - 2.59 = 27.33$ inches. The lower the temperature to which the condensed steam is reduced the more nearly perfect will be the vacuum attained.

If air is mixed with the steam the vacuum will be still more imperfect. Thus, suppose the vessel to contain one pound of steam and one-tenth of a pound of air under atmospheric pressure. The volume of the closed vessel in this case must be $26.79 + 1.69 = 28.48$ cubic feet. (1.69 = volume of $\frac{1}{10}$ pound of dry air at 212 degrees F. and 29.92 inches of mercury pressure.)

After the steam has been condensed and its temperature reduced to 110 degrees F. the absolute pressure in the condenser will be 4.1 inches, thus:

According to Dalton's Laws: (1) The mass of a given kind of vapor required to saturate a given space at a given temperature is the same whether the vapor is all by itself or associated with vaporless gases; (2) the maximum tension of a given kind of vapor at a given temperature is the same whether it is all by itself or associated with vaporless gases; (3) in a mixture of gas and vapor the total pressure is equal to the sum of the partial pressures. The final pressure P_c in the vessel is therefore the combined pressure of the air P_a and that of the water vapor P_v, or, assuming complete saturation,

$$P_c = P_a + P_v \qquad (163)$$

and the final volume of the entrained air V_a will be that of the vessel, which in the specific case under consideration is $V_a = V_c = 28.48$ cubic feet.

The final pressure of the air in the vessel may be calculated from the well-known physical law

$$\frac{P_1 V_1}{T_1} = \frac{P_a V_a}{T_a}, \tag{164}$$

in which

P_1 and P_a = absolute pressures corresponding to absolute temperatures T_1 and T_a;

V_1 and V_a = volumes corresponding to absolute temperatures T_1 and T_a.

Here $P_1 = 29.92 \quad V_1 = 1.69 \quad T_1 = 460 + 212 = 672,$
$V_a = 28.48 \quad T_a = 460 + 110 = 570.$

Substitute these values in equation (164)

$$\frac{29.92 \times 1.69}{672} = \frac{P_a \times 28.48}{570},$$

from which $P_a = 1.51$.

The final pressure in the vessel is, equation (163),

$$P_c = 1.51 + 2.59 = 4.1 \text{ inches of mercury.}$$

Example: If the absolute pressure in a condenser is 4 inches of mercury and the temperature of the air-vapor mixture is 100 degrees F., required the percentage of air by weight in the mixture.

From steam tables the pressure of vapor corresponding to a temperature of 100 degrees F. is 1.93 inches of mercury.

Hence, from equation (163),

$$P_c = P_a + P_v,$$
$$4 = P_a + 1.93,$$
$$P_a = 2.07.$$

Let V = volume of the condenser chamber, cubic feet.

Then $0.00285\ V$ = weight of vapor in the chamber (0.00285 = density of water vapor at 100 degrees F.), and

$0.08635 \times \frac{2.07}{29.92} \times \frac{460 + 0}{460 + 100} V = 0.00491\ V$ = weight of dry air in the chamber. (0.08635 = density of air at 0 degrees F. and 29.92 inches of mercury pressure.)

The total weight of the mixture is

$$0.00285\ V + 0.00491\ V = 0.00776\ V,$$

and the percentage of air in the mixture is

$$\frac{0.00491\ V}{0.00776\ V} = 0.632 \text{ or } 63.2 \text{ per cent.}$$

In practice air is always present in exhaust steam. A condenser is a device in which the process of condensation and subsequent removal of the air and condensed steam is continuous, the degree of vacuum obtained depending upon the tightness of valves and joints, the quantity of entrained air, and the temperature to which the condensed steam is reduced.

The degree of vacuum may be expressed in different ways. (1) Excess of the atmospheric pressure over the observed vacuum. For example, a 26-inch vacuum implies that the pressure of the atmosphere is 26 inches of mercury above the pressure in the condenser. (2) Per cent of vacuum, by which is meant the ratio of the observed vacuum to the atmospheric pressure. Thus, with the barometer standing at 30 inches, a vacuum of 26 inches may be expressed as $100 \times \frac{26}{30} = 86.6$ per cent vacuum. This method of expression gives an idea of the efficiency of the condensing system. For example, the degree of vacuum indicated by 26 inches would be 93 per cent with a barometric pressure of 28 inches but only 84 per cent when the barometer reads 31 inches. (3) Absolute pressure. Thus a 26-inch vacuum referred to a 30-inch barometer would be indicated as a pressure of $30 - 26 = 4$ inches absolute, or 1.99 pounds per square inch.

The mean atmospheric pressure at sea level is 14.7 pounds per square inch, corresponding to a mercury column 29.92 inches in height, temperature of the mercury 32 degrees F. If the reading of the vacuum gauge and of the barometer are both corrected to 32 degrees F., the difference gives the absolute pressure in inches of mercury. This is the usual method in average scientific investigations. In condenser practice it is customary to refer the readings of the vacuum gauge to a 30-inch barometer in which case it is necessary to increase the standard temperature of the mercury to such a figure as will increase the height of the mean barometer from 29.92 to 30 inches, viz., 58.4 degrees F. Thus, if the barometer and the vacuum gauge readings are corrected to a temperature of 58.4 degrees F., the difference between the figures will give the absolute pressure in inches of mercury at 58.4 degrees F., and if the figure is subtracted from 30 inches, it will give the inches of vacuum referred to a standard barometer of 30 inches.

The mercury column correction for any change in temperature may be closely approximated by the equation

$$h = h_1 [1 - 0.000101 (t_1 - t)],$$

in which

h = height of mercury column corrected to temperature t;
h_1 = observed height of mercury column;
t_1 = observed temperature of mercury column;
t = temperature to which column is to be referred.

Example: Height of mercury in vacuum gauge 28.52 inches, temperature of mercury 80 degrees F., barometer 29.85 inches, temperature 42 degrees F.; determine the vacuum referred to a 30-inch barometer.

For the vacuum gauge

$$\begin{aligned} h &= 28.52\,[1 - 0.000101\,(80 - 58.4)] \\ &= 28.46. \end{aligned}$$

For the barometer

$$\begin{aligned} h &= 29.85\,[1 - 0.000101\,(42 - 58.4)] \\ &= 29.9. \end{aligned}$$

Absolute pressure in inches of mercury at temperature 58.4 degrees F. = 29.9 − 28.46 = 1.44.

Vacuum referred to 30-inch barometer = 30 − 1.44 = 28.56.

Properties of Air and Steam Mixtures in Relation to Condensing Plant: Engng., Jan. 19, 1912.

The Influence of Air on Vacuum in Surface Condensers: Engng., Apr. 17, 1908; Power, Feb. 2, 1909, p. 235; Nov. 22, 1910; Jour. A. S. M. E., Nov., 1912.

Properties of Dry, Saturated and Unsaturated Air: Jour. Frank. Inst., Feb., 1911.

240. Function of the Condenser. — The function of a condenser in connection with a steam engine or turbine is primarily the reduction of back pressure, though, in some instances, notably in marine work, the recovery of the condensed steam may be of equal importance. The advantages to be gained by decreasing back pressure may be most readily illustrated by the following example: A non-condensing engine taking steam at a pressure of 100 pounds absolute and cutting off at one-quarter stroke will have, theoretically, a mean effective pressure on the piston of 44.6 pounds per square inch, the back pressure being 14.7 pounds per square inch absolute. If the engine exhausts into a condenser against a 26.5-inch vacuum (1.7 pounds absolute) the mean effective pressure will be increased to 44.6 + (14.7 − 1.7) = 57.6 pounds per square inch, resulting in a gain in power which may be expressed

$$\text{H.P.} = \frac{P_r A S}{33{,}000}, \tag{165}$$

in which

H.P. = horse power gained;
P_r = reduction in back pressure, pounds per square inch;
A = area of the piston in square inches;
S = piston speed in feet per minute.

If P = mean effective pressure on the piston when running non-condensing, the percentage of increase of power may be expressed

$$\text{Per cent} = 100\,\frac{P_r}{P}. \tag{166}$$

In the above example the percentage of power gained would be

$$100 \frac{13}{44.6} = 29.2 \text{ per cent.}$$

The actual gain due to the use of the condenser would be much less than this, depending upon the type of engine and conditions of operation, as shown in the results of engine performances outlined in Chapter X.

TABLE 78.

PRESSURE OF AQUEOUS VAPOR IN INCHES OF MERCURY FOR EACH DEGREE F.

(Marks and Davis.)

	0°	1°	2°	3°	4°	5°	6°	7°	8°	9°
30°.........	......	......	.180	.188	.195	.203	.212	.220	.229	.238
40°.........	.248	.257	.268	.278	.289	.300	.312	.324	.336	.349
50°.........	.362	.376	.390	.405	.420	.436	.452	.468	.486	.503
60°.........	.522	.541	.560	.580	.601	.622	.644	.667	.690	.714
70°.........	.739	.764	.790	.817	.845	.873	.903	.964	.996	1.03
80°.........	1.03	1.06	1.10	1.13	1.17	1.21	1.25	1.30	1.33	1.37
90°.........	1.42	1.46	1.51	1.55	1.60	1.65	1.71	1.76	1.81	1.87
100°.........	1.93	1.98	2.04	2.11	2.17	2.24	2.30	2.37	2.44	2.51
110°.........	2.60	2.66	2.74	2.82	2.90	2.99	3.07	3.16	3.25	3.34
120°.........	3.44	3.53	3.63	3.74	3.84	3.95	4.06	4.17	4.28	4.40
130°.........	4.52	4.64	4.76	4.89	5.02	5.16	5.29	5.43	5.58	5.73
140°.........	5.88	6.03	6.18	6.34	6.51	6.67	6.84	7.02	7.20	7.38

With steam turbines the advantage gained by reduction of back pressure is more marked than with the reciprocating engine, though theoretically the same for the same range of expansion. Initial condensation, leakage past valves, and other sources of loss prevent a reciprocating engine from benefiting from a good vacuum to the same extent as a turbine.

Referring again to the example given above, if the steam is cut off at about one-sixth stroke, the work done when running condensing will be the same as when running non-condensing and cutting off at one-quarter. Theoretically the steam consumption will be decreased nearly in proportion to the reduction in cut-off. Generally speaking, a condensing engine will require from 20 to 30 per cent less steam for a given power than a non-condensing engine. (See results of engine tests, paragraph 206.) This decrease in steam consumption is only an apparent one. If steam is used by the auxiliaries in creating the vacuum, the amount must be added to that consumed by the engine, unless the steam exhausted by the former is utilized to warm the feed water, in which case only the difference between the heat entering the auxiliaries and that returned to the heater should be charged against the engine. The power

necessary to operate the condenser auxiliaries varies from one to six per cent of the main engine power, depending upon the type and conditions of operation. (See paragraph 260.)

In power plants where the exhaust steam is not used for heating or manufacturing purposes, the engines are almost invariably operated condensing, provided there is an abundant supply of cooling water. Even if the water supply is limited, it is often found to be economical to use some artificial cooling device, notwithstanding the high first cost and cost of operation of the latter.

Some of the considerations affecting the propriety of running condensing and the choice of condensing systems are taken up in paragraphs 263 and 264.

241. Classification of Condensers. — The following is a classification of a few well-known condensers:

1. Jet condensers	Parallel current (*a*)	Standard low Vacuum	Worthington. Blake. Deane.
		Siphon	Baragwanath. Bulkley.
		Ejector	Schutte. Körting.
	Counter current (*b*)	Barometric	Weiss. Alberger.
		High vacuum	Leblanc. Wheeler. Worthington.
2. Surface condensers	Water cooled (*a*)	Single-flow	Baragwanath.
		Double-flow	Wheeler.
		Multi-flow	Wainwright.
	Air cooled (*b*)	Forced draft	Fouche.
		Natural draft	Pennell.
	Evaporative (*c*)		Ledward.

Condensers may be divided into two general groups:

1. *Jet condensers,* in which the steam and cooling water mingle and the steam is condensed by direct contact, Figs. 296 to 303.

2. *Surface condensers,* in which the steam and cooling medium are in separate chambers and the heat is abstracted from the steam by conduction, Figs. 308 to 310.

Jet condensers may be further grouped into two classes, according to the direction of flow of the air and cooling water:

(*a*) *Parallel-current condensers,* in which the condensed steam, cooling water, and air flow in the same direction, collect at the bottom of the condenser chamber, and are exhausted by a suitable pump, Fig. 296.

(*b*) *Counter-current condensers,* in which the cooling water and condensed steam flow from the bottom of the chamber, while the air is drawn off at the top, Fig. 302.

Parallel-current condensers may be subdivided into three classes:

(1) *Standard condensers,* in which the cooling water, condensed steam, and air are exhausted by a vacuum pump, Fig. 296.

(2) *Siphon condensers,* in which the cooling water, condensed steam, and air are exhausted by a barometric column, Fig. 299.

(3) *Ejector condensers,* in which the condensed steam and air are exhausted by the cooling water on the ejector principle, Fig. 300.

Surface condensers may be classified according to the nature of the cooling medium as

(*a*) *Water-cooled condensers.*

(*b*) *Air-cooled condensers.*

(*c*) *Evaporative condensers,* in which the condensation of the steam is brought about by the evaporation of a fine stream of water trickling on the outside of the tubes.

242. Standard Low-vacuum Jet Condensers. — Fig. 296 shows a section through a Worthington jet condenser, illustrating the parallel-current principle. When the pump is started a partial vacuum is created in the suction chamber above the valves *H*, *H* in the cone *F*. As soon as sufficient air has been exhausted, cooling water enters at *B* with a velocity depending upon the degree of vacuum in chamber *F* and the suction head, and is divided into a fine spray by the adjustable serrated cone *D*. The spray mingles with the exhaust steam entering at *A* and both move downwards with diverse velocities. The steam gives up its heat to the water and condenses. The velocity of the steam diminishes in its downward path to zero, while the velocity of the water increases according to the laws of falling bodies. The condensed steam, cooling water, and air collect at the lower part of the condenser and are exhausted by the *wet air pump G*, from which they are forced through opening *J* to the hot well. The vacuum in chamber *F* will depend upon the vapor tension of the warm water in the bottom of the well, the amount of air carried along by the cooling water and steam, and the tightness of valves and joints. In case the water accumulates in the condenser cone *F*, either by reason of an increased supply or by a sluggishness or even stoppage of the pump, the condensing surface is reduced to a minimum, as soon as the level of the water reaches the spray pipe and the spray becomes submerged, and only a small annular surface of water is exposed to the exhaust steam. The vacuum is immediately broken, and the exhaust steam escapes by blowing through the injection pipe and through the valves of the pump and out the discharge pipe at *J*, forcing the water ahead of it; consequently flooding of the steam cylinder cannot occur. In starting up the condenser a partial

vacuum for inducing a flow of injection water into the condenser chamber may be created by the pump if the suction lift is not too great. Many engineers, however, prefer to install a small forced injection or priming pipe the function of which is to condense sufficient steam to produce the necessary partial vacuum.

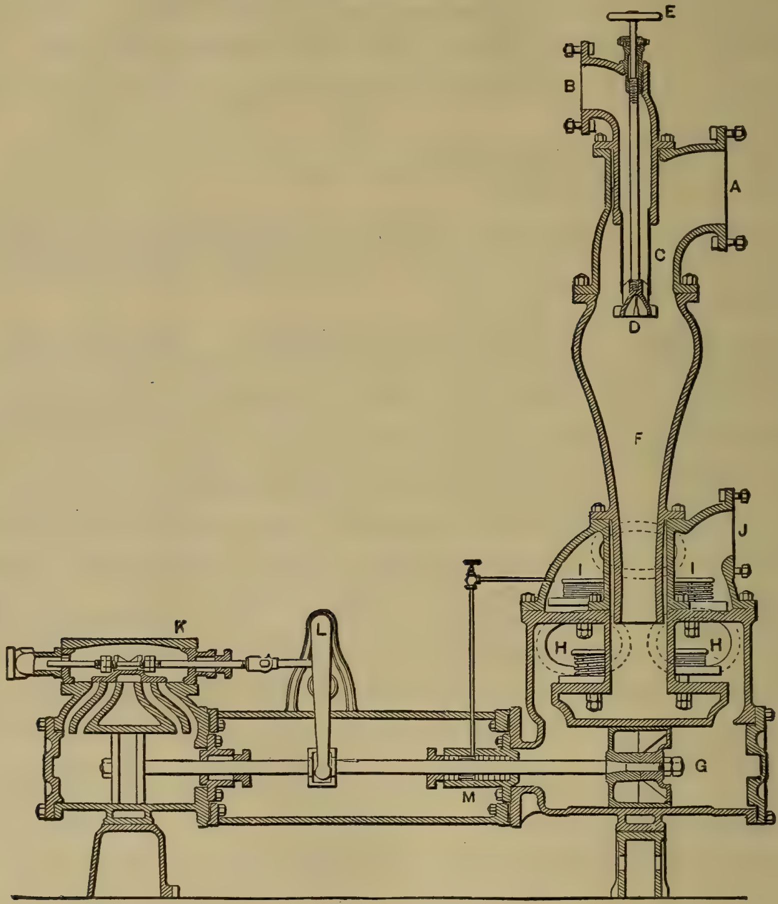


FIG. 296. Worthington Independent Jet Condenser.

Fig. 297 shows a section through the condensing chamber and air pump of a Blake vertical jet condenser with an automatic vacuum-breaking device. The injection water enters at opening marked "injection" and flows through the adjustable "spray" nozzle in a fine spray, at an angle of about 45 degrees, and impinges on the conical sides of the upper condenser chamber. The spray falls from the sides to the projecting ledges shown in the illustration. The ledges prevent the spray

from falling directly to the bottom of the chamber and insure an efficient mingling of steam and cooling water. A perforated copper plate is substituted for the shelves when the force of the injection water is not sufficient to produce spray. The circulating water and condensed steam together with the non-condensable gases are drawn off at the bottom of the chamber. The vacuum-breaking device is shown at the right of the figure. When the rising water reaches the level of the float

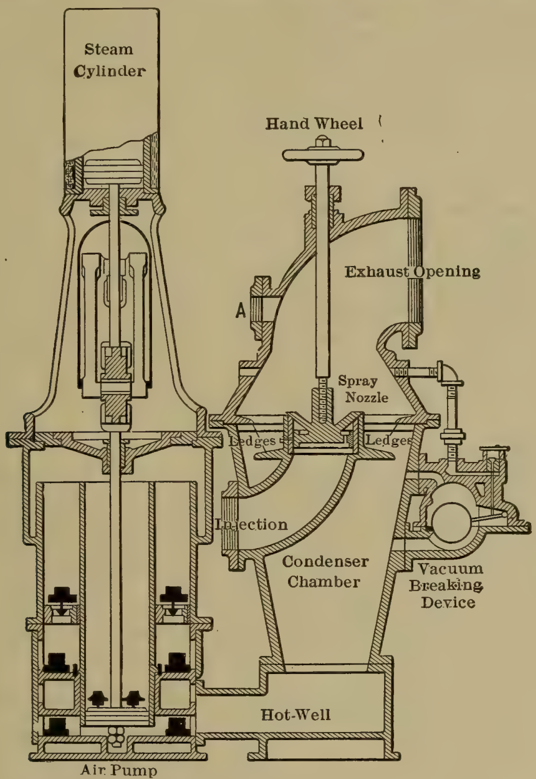


FIG. 297. Section through a Blake Jet Condenser.

chamber, as in the case of an accidental stoppage of the air pumps, the float is raised and forces a check valve from its seat and allows an inrush of air to break the vacuum, thus preventing further suction of water into the condenser and consequent flooding of the engine. *A* is the forced injection or "priming" inlet used in starting up when the suction lift is considerable.

Condensation of Steam: Cassier's Mag., May, 1912; June, 1911.

Jet versus Surface Condensers: Power, March 21, 1911, p. 443; Engr., Lond., Dec. 23, 1910.

Condensers: Power, March 19, 1912.

243. Condensing Water, Jet Condensers. — In a jet condenser the cooling water and exhaust steam mingle, and the degree of vacuum is a function of the final or discharge temperature; thus the quantity of cooling water required depends upon its initial temperature, the temperature of the discharge water, and the total heat in the steam entering the condenser. If the steam in the low-pressure cylinder at exhaust is dry and saturated, the heat entering the condenser will correspond to the total heat in steam at exhaust pressure, but it usually contains considerable moisture, part of which is reëvaporated when the exhaust valve opens to the condenser; however, it is sufficiently accurate for most practical purposes to assume the exhaust steam entering the condenser to be dry and saturated and its heat to correspond to the pressure in the condenser.

Let H = heat content of the steam at condenser pressure,
t_2 = temperature of the discharge water,
t_0 = initial temperature of the cooling water,
W = weight of cooling water in pounds necessary to condense and cool one pound of steam to the required discharge temperature.

Then
$$W = \frac{H - t_2 + 32}{t_2 - t_0}.* \tag{167}$$

Example: How many pounds of cooling water are necessary to condense one pound of steam under the following conditions: Barometer 29.92; vacuum 26 inches; temperature of injection water 60 degrees F.

The temperature of aqueous vapor corresponding to an absolute pressure of $29.92 - 26 = 3.92$ inches of mercury is 125 degrees F. (See Table 78.) The discharge temperature, however, must be less than this, as the pressure in the condenser is due not only to the aqueous vapor but to that of the air carried over with the circulating water and the condensed steam. In a condenser of the standard low-vacuum type the discharge temperature will be from 10 degrees to 15 degrees lower than that corresponding to the vacuum as recorded by the gauge. In this case assume it to be 15 degrees lower, i.e., $t_2 = 125 - 15 = 110$ degrees.

The total heat corresponding to a pressure of 3.92 inches of mercury is 1114 B.t.u. above 32 degrees (see steam tables); $t_0 = 60$ degrees; $t_2 = 110$ degrees.

$$W = \frac{1114 - 110 + 32}{110 - 60} = 20.7.$$

* This expression is not theoretically correct, since it assumes a constant value of unity for the mean specific heat of water. The variation, however, is so slight that it may be neglected for all practical purposes.

Evidently the higher the temperature of the discharge water the less will be the quantity of cooling water required, and consequently the smaller the weight of air introduced into the condenser; but the warmer the discharge water the greater will be the vapor tension and the lower

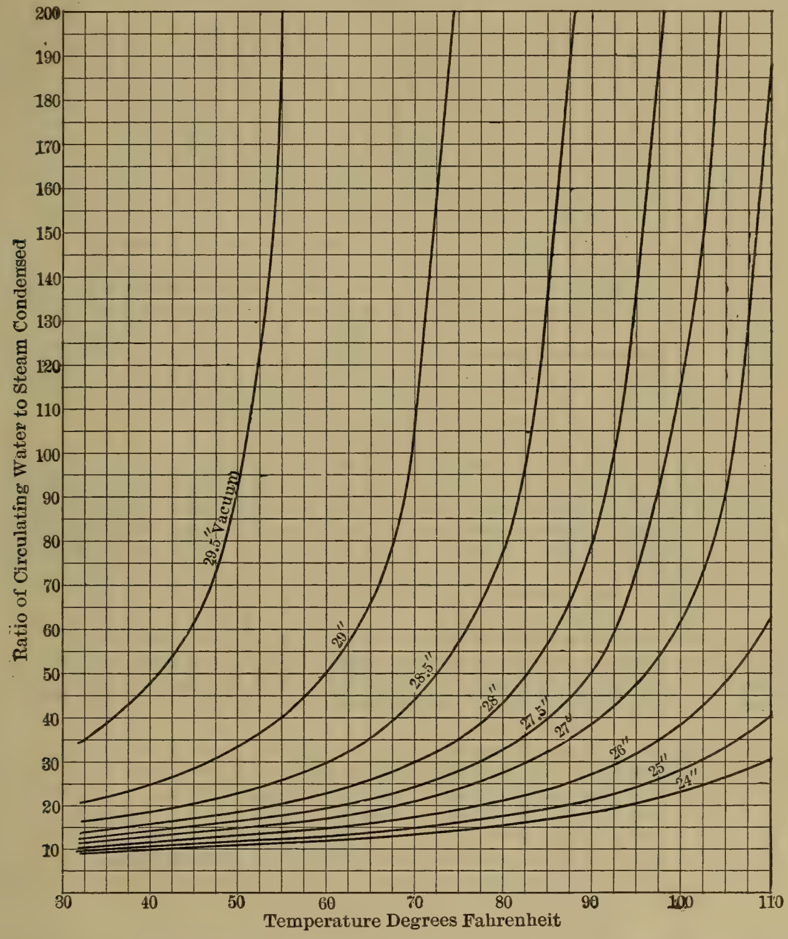


FIG. 298. Curves showing Relation Between Cooling Water and Hot-well Temperatures.

the degree of vacuum. For reciprocating engines a hot-well temperature between 110 degrees and 130 degrees F. is average practice; with turbines the temperature ranges between 80 degrees and 100 degrees F. On account of the inefficient heat absorption in practical installations, from 5 per cent to 15 per cent is added to the theoretical weight of cooling water as determined from equation (167). With jet condensers of the Leblanc type the temperature of the hot well is approximately

TABLE 79.

RATIO, BY WEIGHT, OF COOLING WATER TO STEAM CONDENSED (THEORETICAL).

(Barometer 29.92.)

Temp. of Injection.	Vacuum 24". Temperature of Steam 141°. Temperature of Hot Well.				
	110	115	120	125	130
40	15.0	13.9	12.9	12.1	11.4
50	17.5	16.0	14.8	13.7	12.8
60	21.0	18.9	17.3	15.8	14.6
70	26.2	23.2	20.7	18.7	17.1
80	35.0	29.8	25.9	23.0	20.5
90	52.4	49.7	34.6	29.5	25.6

Temp. of Injection.	Vacuum 25". Temperature of Steam 134°. Temperature of Hot Well.				
	105	110	115	120	125
40	16.1	14.9	13.8	12.9	12.1
50	19.0	17.4	16.0	14.8	13.7
60	23.2	20.9	18.9	17.2	15.8
70	30.0	26.1	23.0	20.7	18.7
80	42.0	34.8	29.6	25.9	22 8
90	70.0	52.1	41.5	34.5	29.4

Temp. of Injection.	Vacuum 26". Temperature of Steam 125°. Temperature of Hot Well.				
	100	105	110	115	
40	17.5	16.1	14.8	13.8	......
50	21.0	19.0	17.4	16.0	......
60	26.3	23.2	20.9	18.8	......
70	35.0	30.0	26.0	23.0	......
80	57.6	42.0	34.7	29.6	......

Temp. of Injection.	Vacuum 27". Temperature of Steam 114°. Temperature of Hot Well.				
	90	95	100	105	
40	21.2	19.1	17.4	16.0	......
50	26.5	23.4	20.9	19.0	......
60	35.3	30.1	26.2	23.2	......
70	52.9	42.1	34.9	29.8	......
80	......	......	52.3	41.5	......

Temp. of Injection.	Vacuum 27.5". Temperature of Steam 108°. Temperature of Hot Well.				
	80	85	90	95	
40	26.6	23.6	21.1	19.1	......
50	35.6	30.3	26.4	23.4	......
60	52.3	42.5	35.2	30.0	......
70	......	70.8	52.8	42.0	......

Temp. of Injection.	Vacuum 28". Temperature of Steam 100°. Temperature of Hot Well.				
	75	80	85	90	
40	30.5	26.6	23.5	21.1	......
50	42.7	35.5	30.2	26.3	......
60	71.2	53.2	42.3	35.1	......
70	......	......	70.6	52.7	......

Temp. of Injection.	Vacuum 28.5". Temperature of Steam 90°. Temperature of Hot Well.				
	60	65	70	75	
35	42.2	35.8	30.6	29.2	......
40	54.0	43.0	35.6	33.4	......
45	72.0	53.5	42.8	38.8	......
50	......	72.0	53.5	46.6	......

Temp. of Injection.	Vacuum 29". Temperature of Steam 77°. Temperature of Hot Well.				
	55	60	65	67	
35	52.0	43.0	35.8	33.4	......
40	69.3	54.0	43.0	38.4	......
45	......	71.5	54.0	47.0	......
50	......	......	72.0	61.0	......

5 degrees lower than that corresponding to the degree of vacuum in the condenser for inlet temperatures above 50 degrees F. For lower temperatures of inlet water the hot-well temperature ranges from 10 to 20 degrees below that corresponding to the degree of vacuum. Table 79 and the curves in Fig. 298 have been calculated from equation (167).

244. Effect of Aqueous Vapor upon the Degree of Vacuum.— The futility of attempting to better the vacuum by exhausting the vapor is best illustrated by a specific problem.

Required the volume of aqueous vapor to be withdrawn per hour from a condenser operating under the following conditions, in order that the vacuum may be increased one pound per square inch: Temperature of discharge water 125 degrees; corresponding vapor tension 4 inches of mercury; barometer 30 inches; relative vacuum 26 inches; horse power 100; steam consumption 20 pounds per horse-power hour; cooling water 25 pounds per pound of steam condensed.

$100 \times 20 \times 25$ = 50,000 pounds of cooling water per hour.
= 833 pounds of cooling water per minute.

Now to increase the vacuum one pound per square inch, approximately 2 inches of mercury, the temperature of the water must be lowered to 102 degrees F., that is, 833 (125 − 102) = 19,159 B.t.u. must be abstracted from the water in one minute, or $\frac{19,159}{1030} = 18.6$ pounds of water to be evaporated per minute. (1030 = average heat of vaporization of water under 26 to 28 inches of vacuum.) Now, one pound of vapor at 102 to 125 degrees F. has an average volume of 270 cubic feet.

Therefore $18.6 \times 270 = 5022$ cubic feet of vapor must be exhausted per minute to increase the vacuum from 26 to 28 inches, which is manifestly impracticable.

245. Injection Orifice. — The velocity of water entering a jet condenser, neglecting friction, may be determined from the formula

$$V = \sqrt{2\,gh}, \tag{168}$$

where

V = velocity of the water in feet per second,
g = acceleration of gravity = 32.2,
h = total head in feet.

If p = pressure below the atmosphere in pounds per square inch,
h_1 = distance in feet between the source of supply and the injection orifice,

then

$$h = 2.3\,p \pm h_1, \tag{169}$$

and equation (168) may be written

$$V = 8.025 \sqrt{2.3\, p \pm h_1}. \tag{170}$$

If the supply is under pressure, h_1 is positive; if under suction, it is negative.

Example: What is the theoretical velocity of water entering a condenser with 26-inch vacuum (referred to 30-inch barometer); suction head 8 feet.

Here p = pressure in pounds per square inch, corresponding to 26 inches of mercury = 12.8 pounds per square inch.

$$h_1 = 8.$$
$$V = 8.025 \sqrt{2.3 \times 12.8 - 8}$$
$$= 37.1 \text{ feet per second}$$
$$= 2226 \text{ feet per minute.}$$

In proportioning the injection orifice in practice the maximum velocity of flow is assumed to be between 1500 and 1800 feet per minute, or, approximately, area of injection orifice in square inches = weight of injection water in pounds ÷ 650 to 780. ("Manual of Marine Engineering," Seaton, p. 204.) A rough rule gives area of orifice = area of low-pressure piston in square inches ÷ 250. (Seaton, p. 204.)

246. Volume of the Condenser Chamber. — According to Thurston the volume of a jet condenser should be from one fourth to one half that of the low-pressure engine cylinder. ("Steam Engine Manual," Thurston, II, 127.)

According to Hutton the volume should not be less than that of the air pump and should approximate three fourths of that of the engine cylinder in communication with it.

247. Injection and Discharge Pipes. — In practice the diameter of the injection pipe is based on a velocity of 400 to 600 feet per minute and that of the discharge pipe of 200 to 400 feet per minute; the lower figures for pipes under 8 inches in diameter, the upper range for larger diameters.

(Atmospheric relief valves. — See paragraph 39.)

248. Siphon Condensers. — Fig. 299 shows a section through a Baragwanath siphon condenser, illustrating the principles of a parallel-current barometric condenser. The cooling water enters the side of the condenser chamber at A and passes downward in a thin annular sheet around the hollow cone D. The exhaust steam enters at B and is given a downward direction by the goose neck C. It flows through the nozzle D and is condensed within the hollow cone of moving water, the combined mass including the entrained air discharging through the contracted throat E at high velocity into the tail pipe F. The water

column in the tail pipe must be enough to overcome the pressure of the atmosphere; i.e., it should be 34 feet or more above the surface of the hot well, otherwise water would rise within this pipe to a height corresponding to that of the barometer, which is approximately 34 feet for a barometric pressure of 30 inches of mercury. This is not strictly true when the condenser is in full operation, as the injector effect of the moving mass is sufficient to overcome several pounds pressure, and the tail pipe may be less than 34 feet, but to provide against any possibility of the water being drawn into the cylinder of the engine the length is made greater than 34 feet. The spray cone *D* is adjustable and admits of close regulation of the water supply without changing the annular form of the stream. The condensing water may be supplied under pressure or under suction. For lifts not greater than 15 feet no supply pump is necessary, the water being raised by the siphon action of the condenser. This condenser requires the same amount of cooling water per pound of steam as the standard jet condenser, and is capable of maintaining a vacuum of from 24 to 27 inches. A vacuum of $28\frac{1}{2}$ inches has been recorded for a condenser of this general type. (Trans. A.S.M.E., 26–388.) An atmospheric relief valve *G* is provided in case the vacuum fails from any cause, which will permit the steam to escape to the atmosphere.

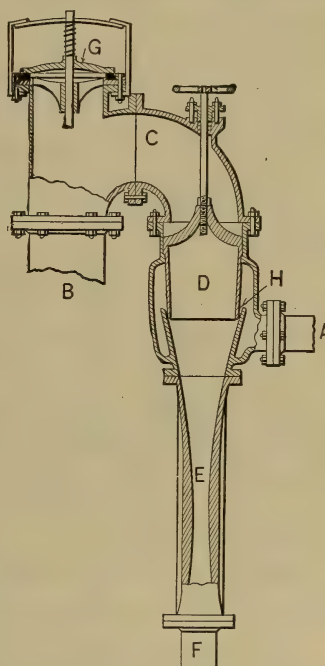


FIG. 299. Baragwanath Siphon Condenser.

The above type of condenser is adapted to very muddy cooling water, since no filtration is necessary beyond the removal of such solid matter as may clog up the annular space *H*.

In the Armour Glue Works at Chicago condensers of this type are successfully maintaining a 90 per cent vacuum with cooling water at 60 degrees F.

Siphon Condensers, Discussion: Trans. A.S.M.E., Vol. 26, p. 388. *Siphon Condensers:* Electrical World, June, 1897, p. 818; Engr. U. S., Jan., 1906.

249. Size of Siphon Condensers. — The size of siphon is indicated by the diameter of the engine exhaust pipe.

Table 80 gives the sizes of barometric condensers as manufactured by prominent makers.

TABLE 80.

SIZE OF SIPHON CONDENSERS.

Steam to be Condensed.		Size Usually Furnished, Inches.	Steam to be Condensed.		Size Usually Furnished, Inches.
Pounds per Hour.	Pounds per Minute.		Pounds per Hour.	Pounds per Minute.	
2,000	33	5	8,000	133	10
3,000	50	7	10,000	166	12
4,000	66	8	15,000	250	14
5,000	83	9	20,000	333	14
6,000	100	9			

Vacuum 26 inches; barometer 30 inches.

The diameter of the throat may be closely approximated by the empirical formula

$$\text{Diam. in inches} = 0.0077\sqrt{Ww}, \qquad (171)$$

in which

W = weight of steam to be condensed per hour,

w = weight of water required to condense one pound of steam.

The maximum width of the annular opening for the admission of water may be obtained from the empirical formula

$$\text{Width in inches} = \frac{Ww}{39{,}550\, d}, \qquad (172)$$

in which

d = diameter of the nozzle or bottom of the cone in inches.

W and w as in equation (171).

250. Ejector Condenser. — Fig. 300 shows a section through a Schutte exhaust steam "induction" condenser, illustrating the principles of the ejector condenser in which the momentum of flowing water ejects the discharge without the aid of the circulating pump. Exhaust steam enters the ejector through the opening marked "exhaust," passes through a series of inclined orifices and nozzles at considerable velocity, and, meeting the cooling water in the inner annular chamber, is condensed. The cooling water is drawn in continuously through the opening marked "water," by virtue of the vacuum formed, and sufficient velocity is imparted to the jet to discharge the combined mass of condensed steam, cooling water, and air against the pressure of the atmosphere.

Adjustment for capacity is effected by raising or lowering the ram R by means of the wheel H. An adjustable sleeve controls the available area of the exhaust inlet by covering more or less openings in the

combining tube. When the cooling water is supplied under pressure the openings marked "steam" and O are blanked. When water is taken under suction and water under pressure is available for starting, O is blanked and opening marked "steam" is connected with the pressure supply. When water is taken under high suction and live steam is used for starting, inlet marked "steam" is connected to live steam and an overflow check valve is placed at O. Fig. 301 gives an outline of the necessary piping for a condenser installation of this type. These condensers are made in all sizes conforming with exhaust pipe diameters of $1\frac{1}{2}$ to 20 inches. The same amount of cooling water is required as for jet condensing and vacua of 20 to 25 inches are readily obtained.

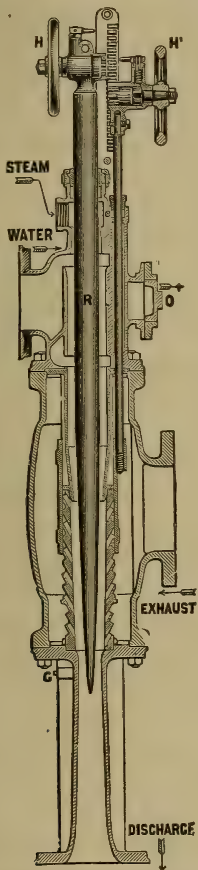


FIG. 300. Schutte Ejector Condenser.

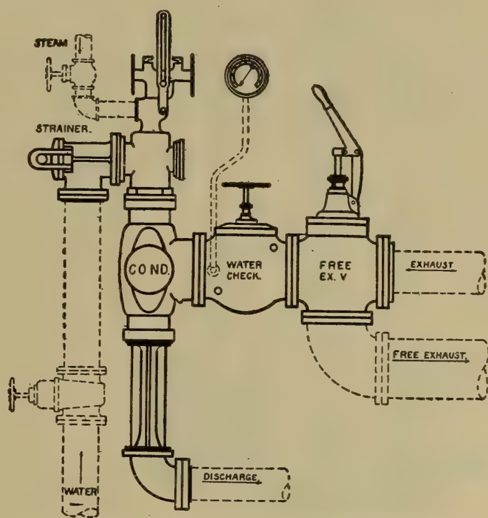


FIG. 301. Piping for Schutte Ejector Condenser.

251. Barometric Condensers.* — Fig. 302 shows a section through a Weiss counter-current condenser, illustrating the principles of a barometric jet condenser. The cooling water enters the upper part of the condensing chamber A through pipe D and falls in cascades, as shown in the figure, to tail pipe B, from which it flows by gravity to the hot well. The exhaust steam enters chamber A through pipe D,

* The author has been informed that the word "Barometric" in connection with jet condensers is the registered trade mark of the Alberger Condenser Company.

and, coming in contact with the cold-water spray, is condensed. The air is exhausted from the top of the condenser by a dry vacuum pump through pipe F. In flowing to the pump the air passes upwards through the water spray and its temperature is lowered to that of the injection water, thereby reducing the volume to be exhausted. Any moisture passing over with the air is separated at G before reaching the air pump, and flows out through the small barometric tube H. The cooling water is forced to the condenser chamber through pipe N by any positive displacement pump, the actual head pumped against being the difference between the total height and that of a column of water corresponding to the degree of vacuum in the condenser. The main barometric tube or tail pipe B through which the water is discharged is 34 feet or more in length and is provided with a foot valve C. The counter-current principle permits a much higher temperature of hot well for the same degree of vacuum than does the parallel current, a hot-well temperature of 120 degrees and a vacuum of 27 inches being readily maintained. A small pipe K connecting the main condenser with the small barometric tube H insures at all times a sufficient quantity of water in the small auxiliary hot well to seal the tube. The water from this auxiliary hot well flows over a weir, as indicated, into a counterweighted bucket M, the latter having a hole in the bottom which allows the normal flow to escape. But in case a sudden heavy overload is thrown on the engines, and the adjustment is for a light load, the temperature of the discharge will reach the boiling point and an abnormal quantity of water will flow down the small barometric tube. This will cause the water to flow into the bucket much faster than the opening in the bottom can dispose of it; as a result the bucket will increase in weight and will open up a free-air valve L which reduces the vacuum two or three inches and raises the boiling point without "dropping" the vacuum entirely. E is the atmospheric relief valve.

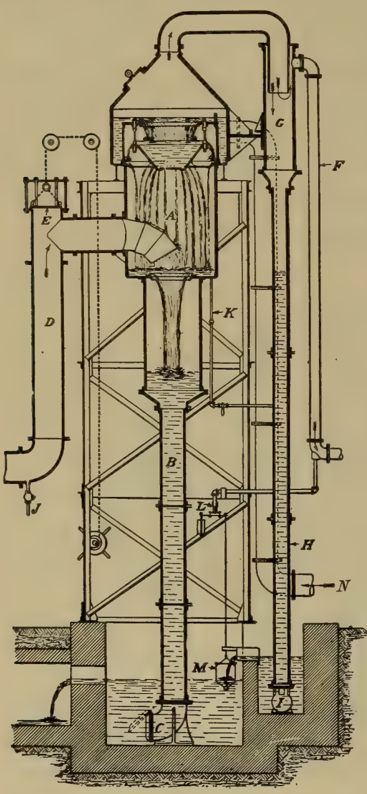


Fig. 302. Weiss Counter-current Condenser.

Fig. 303 shows a section through the condensing chamber of an Alberger barometric condenser. In principles of operation the condenser is similar to the Weiss, but differs considerably in details. Exhaust steam enters at *A* and divides into two streams, one flowing directly to the inner chamber *D*, the other through the annular space *E*. Cooling water enters through *B* and is broken up into a fine spray by the serrated cone *F*, which is hung upon a long spring, thus automatically adjusting itself to the quantity of water entering the condenser. After condensing the exhaust steam in the inner cylinder the partly heated spray of cooling water in falling is brought in contact with the exhaust steam which enters through the annular space. This process permits of a high hot-well temperature without affecting the degree of vacuum. The air which is not entrained by the cooling water and carried down the tail pipe collects under the spray cone *F* and ascends through the tubular support of the cone into the air cooler. This air cooler is simply a small chamber in which the non-condensable gases are cooled by a small portion of the circulating water before they are withdrawn by the air pump. The circulating water used for the purpose is forced into the cooling chamber through pipe *K* and falls through serrated openings in the bottom to the condenser proper. The air enters the chamber through these same openings, and is withdrawn by the air pump. Surrounding the cooler is a separating space of large capacity to allow the subsidence of any entrained moisture before the air reaches the vacuum pump.

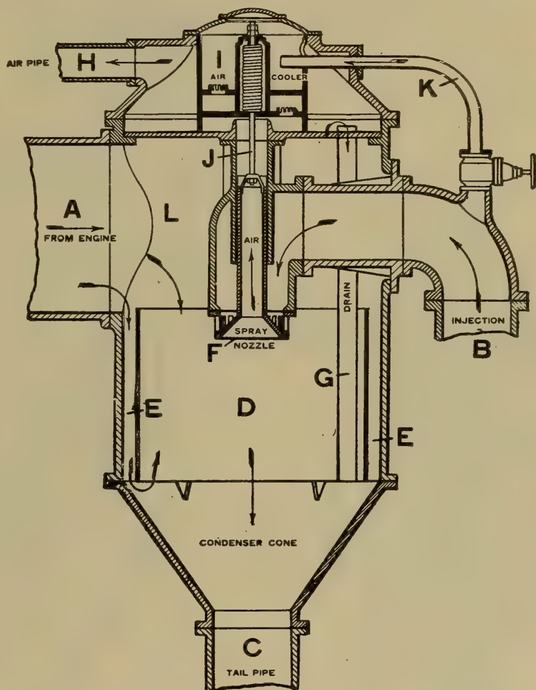


Fig. 303. Section through Condensing Chamber, Alberger Barometric Condenser.

Fig. 304 shows a section through a Tomlinson type B barometric condenser which differs from the conventional type in the addition of an

overflow or auxiliary tail pipe. The main tail pipe takes care of the light loads and the overflow comes into service only on full loads and overloads. This arrangement reduces the quantity of circulating water required at light loads since it is not necessary to keep a large tail pipe filled with water as is the case with the single pipe design.

Fig. 305 shows a section through a Worthington counter-current condensing chamber when overhead room is not restricted, and Fig. 306

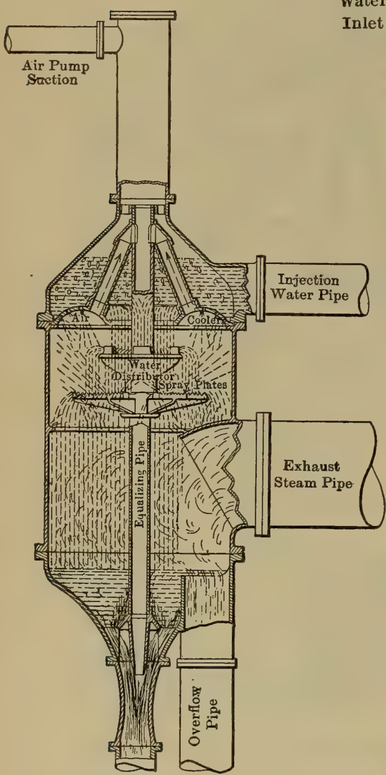


FIG. 304. Tomlinson Type B Barometric Condenser.

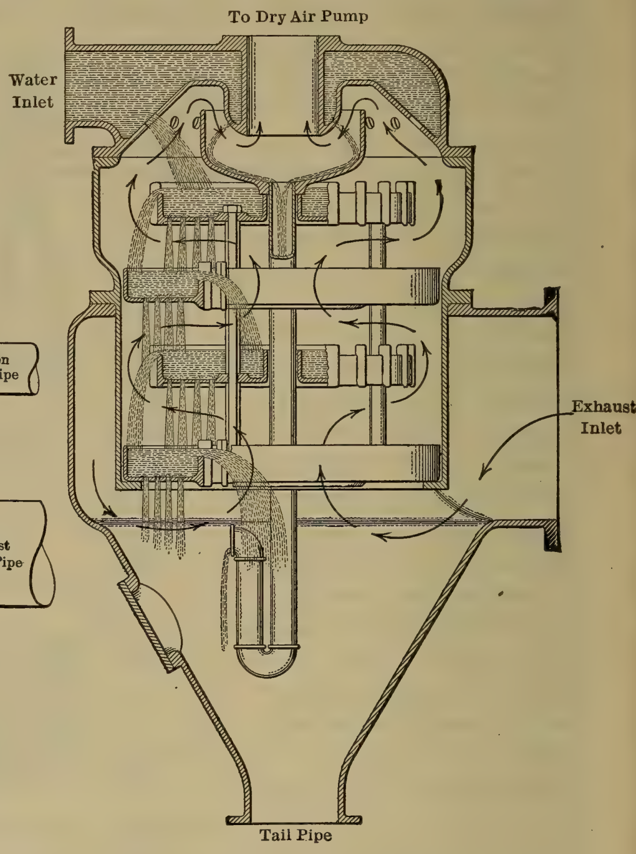


FIG. 305. Worthington Counter-current Jet Condenser.

shows a section through the condensing chamber of a Wheeler low-head condenser.

As previously outlined, surface condensers may be divided into three general classes, (*a*) water-cooled, (*b*) air-cooled, and (*c*) evaporative.

252. Water-cooled Surface Condensers. — Water-cooled surface condensers are by far the most extensive in use and only occasionally are the conditions such as to warrant the installation of the other class.

They are ordinarily classified as (1) single-flow, (2) double-flow, and (3) multi-flow.

Fig. 308 shows a sectional elevation through a Baragwanath vertical condenser, illustrating the single-flow type. It consists essentially of a cast-iron shell provided with two heads, into which a number of one-inch brass tubes are expanded. Exhaust steam fills the shell and flows around and between the tubes, while the cooling water is caused to circulate through the tubes by means of a circulating pump. The steam is condensed by contact with the tubes and drops to the bottom

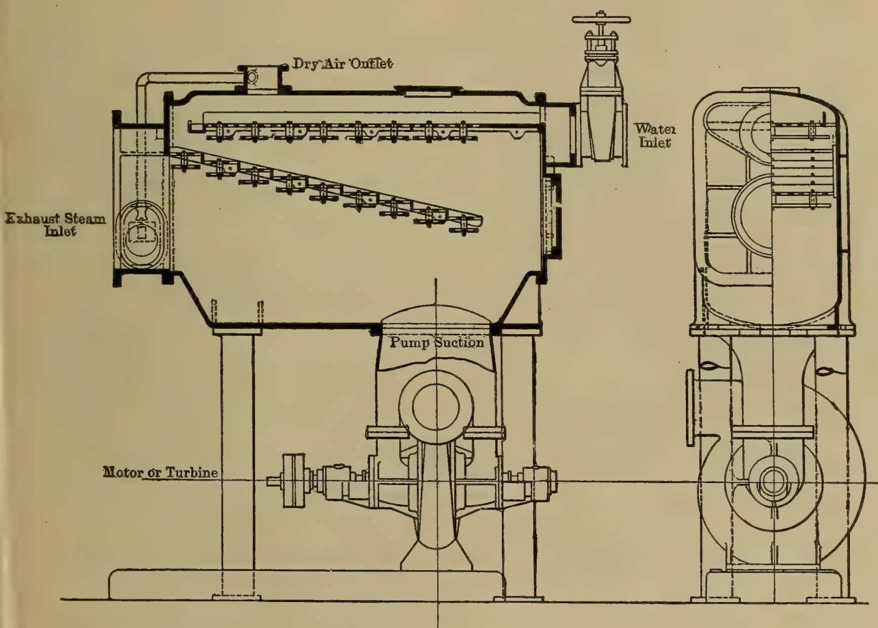


FIG. 306. Wheeler Low-head Centrifugal Jet Condenser.

tube sheet, from which it is exhausted by the air pump. The circulating water flows through the tubes in one direction only, hence the name "single flow." To allow for the unequal expansion of shell and tubes the two halves of the shell are provided with slightly thinner plates flanged outward, the flanges being bolted together with a spacing ring between them. This joint gives to the shell, in the direction of its length, a certain amount of elasticity which is sufficient to allow for the greatest possible elongation of the tubes without straining the tube ends and causing leakage.

Fig. 309 shows a section through a Wheeler admiralty surface condenser mounted on a combined air and circulating pump, illustrating

the typical "double-flow" surface condenser. The condenser proper consists of a ribbed cast-iron chamber of rectangular section fitted with a number of small seamless drawn brass tubes through which the cooling water is forced by suitable means. The exhaust steam enters at the top and is prevented from impinging directly against the tubes by baffle plates, which serve also to distribute the steam more evenly over the cooling surface. The steam in passing between the tubes is condensed, and falls to the bottom of the chamber, from which it is removed, together with the entrained air, by a vacuum pump. The water chamber between the tube sheet and the head is divided into two compartments, as shown in the illustration, the partition being so arranged that the water flows first through the lower set of tubes and then through the upper set in the opposite direction. Thus the temperature of the cooling water increases as it rises, and reaches a maximum where the exhaust steam enters. Condensation begins as soon as the vapor enters the condenser, and the surfaces of the tubes are at once covered with a thin film of water flowing downwards from tube to tube.

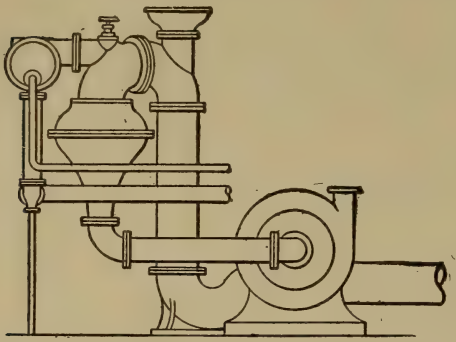


FIG. 307. General Assembly, Centrifugal Jet Condenser System.

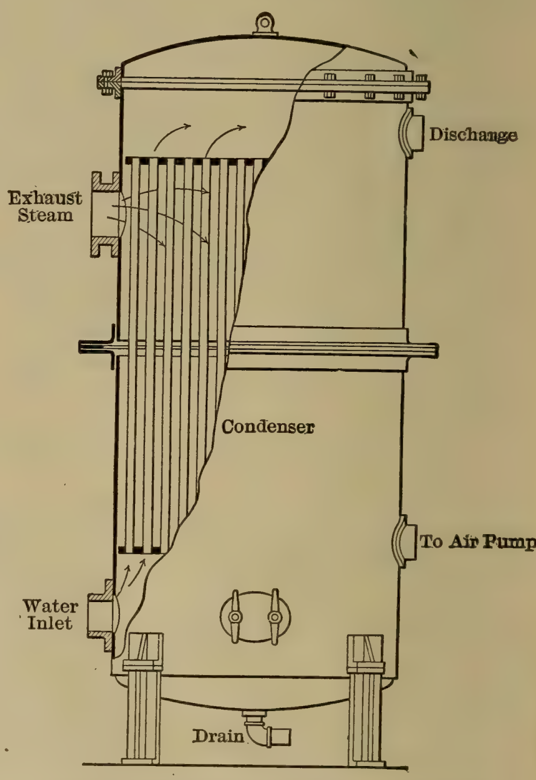


FIG. 308. Baragwanath Surface Condenser.

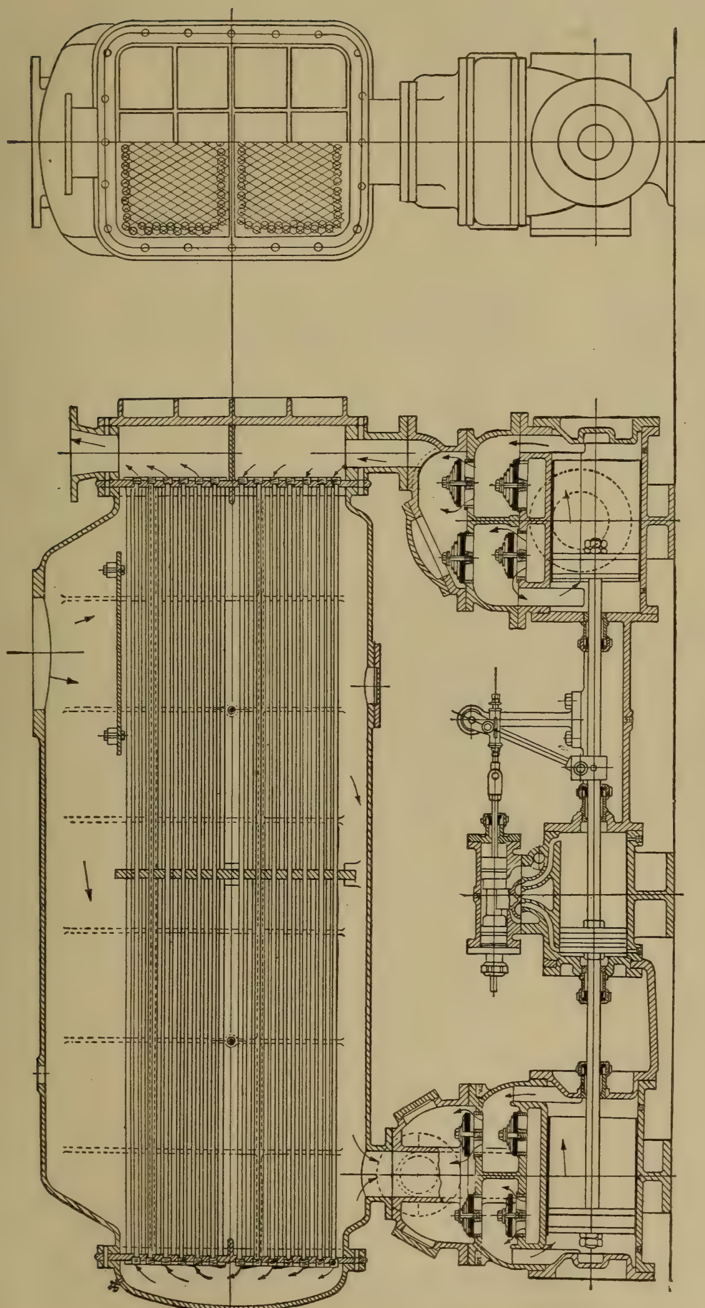


Fig. 309. Wheeler Surface Condenser and Pumps.

Fig. 310 gives the details of a C. H. Wheeler & Company's surface condenser. The condensing chamber is of the series-parallel type in which the water enters the top group of tubes, then passes to the middle section and finally through the bottom section. Connecting chambers are provided at the ends of the shell as illustrated. This construction of water chamber keeps the condenser completely filled with cooling water at all times. The inlet is at the bottom but the water is carried up through the annular chamber to the top of the tubes.

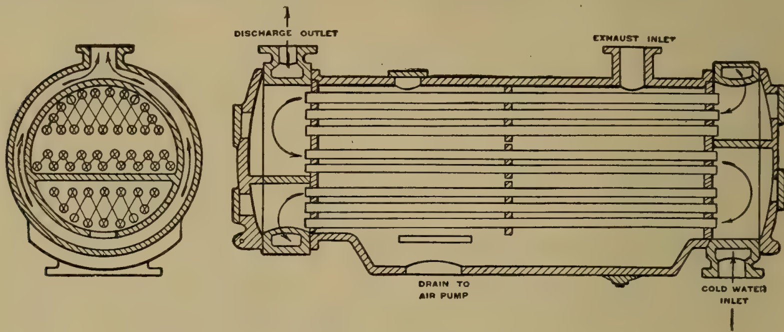


Fig. 310. Surface-Condenser, C. H. Wheeler & Co.

253. High-vacuum Systems. — The average reciprocating engine gives its best commercial economy at a vacuum of approximately 26 inches (referred to a 30-inch barometer), and the ordinary standard jet or surface condenser has been designed to meet this requirement. At the time of the introduction of the steam turbine it was discovered that a very high vacuum would improve turbine economies to an extent hitherto impossible when applied to reciprocating engines. This condition naturally created an era of development among the condenser designers. It became evident at once that the old types that were capable of creating a 26-inch or 27-inch vacuum would require considerable modification to maintain a vacuum of 28 inches or 29 inches. The principal improvement has been in the design of the vacuum pumps.

Surface Condensers. — In the older types of surface condensers the water of condensation from the upper tubes is permitted to fall on the rows immediately below, thereby enveloping them with a blanket of water. This greatly reduces the heat transmission and necessitates comparatively low hot-well temperatures for a given degree of vacuum. By inserting baffles, or *rain plates,* between the banks of tubes and separately draining each compartment it is possible to greatly increase the heat transmission and insure high hot-well temperatures. Prof. R. L. Weighton was the first to apply this system. (See The Efficiency of Surface Condensers, Proc. Inst. of Naval Architects, March, 1906.)

Fig. 319 shows a section through a Worthington *dry-tube* surface condenser embodying Professor Weighton's principles.

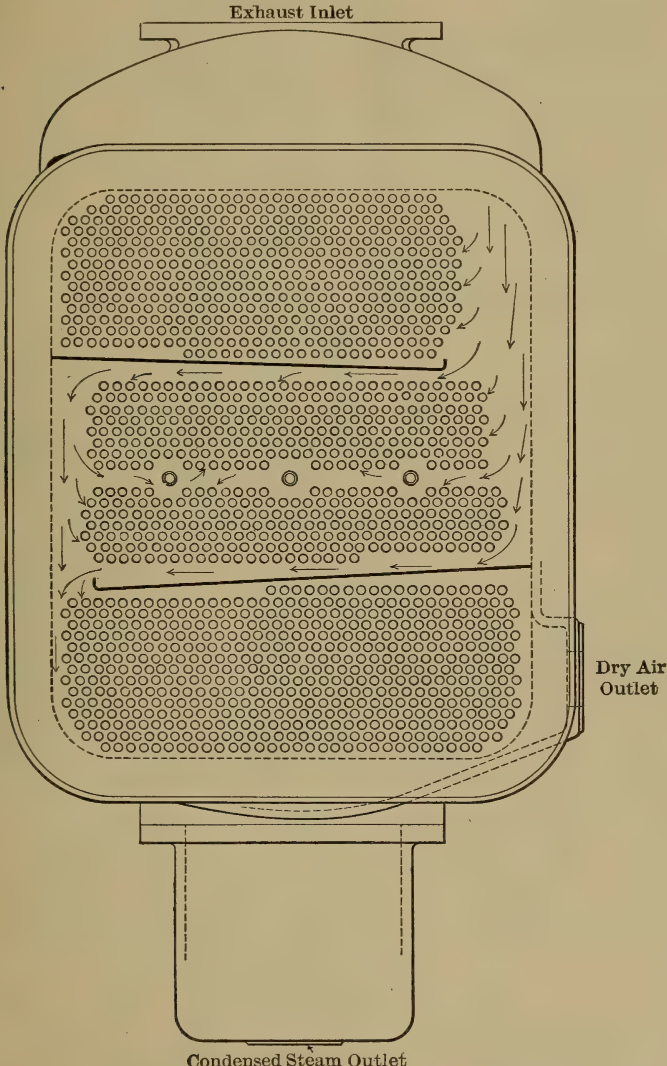


FIG. 311. Wheeler Dry-tube Surface Condenser.

Fig. 312 shows the general arrangement of the Worthington high-vacuum system. The equipment comprises a surface condenser, a steam-driven centrifugal pump for circulating the cooling water, a steam-driven rotative dry-air pump and a turbine-driven centrifugal

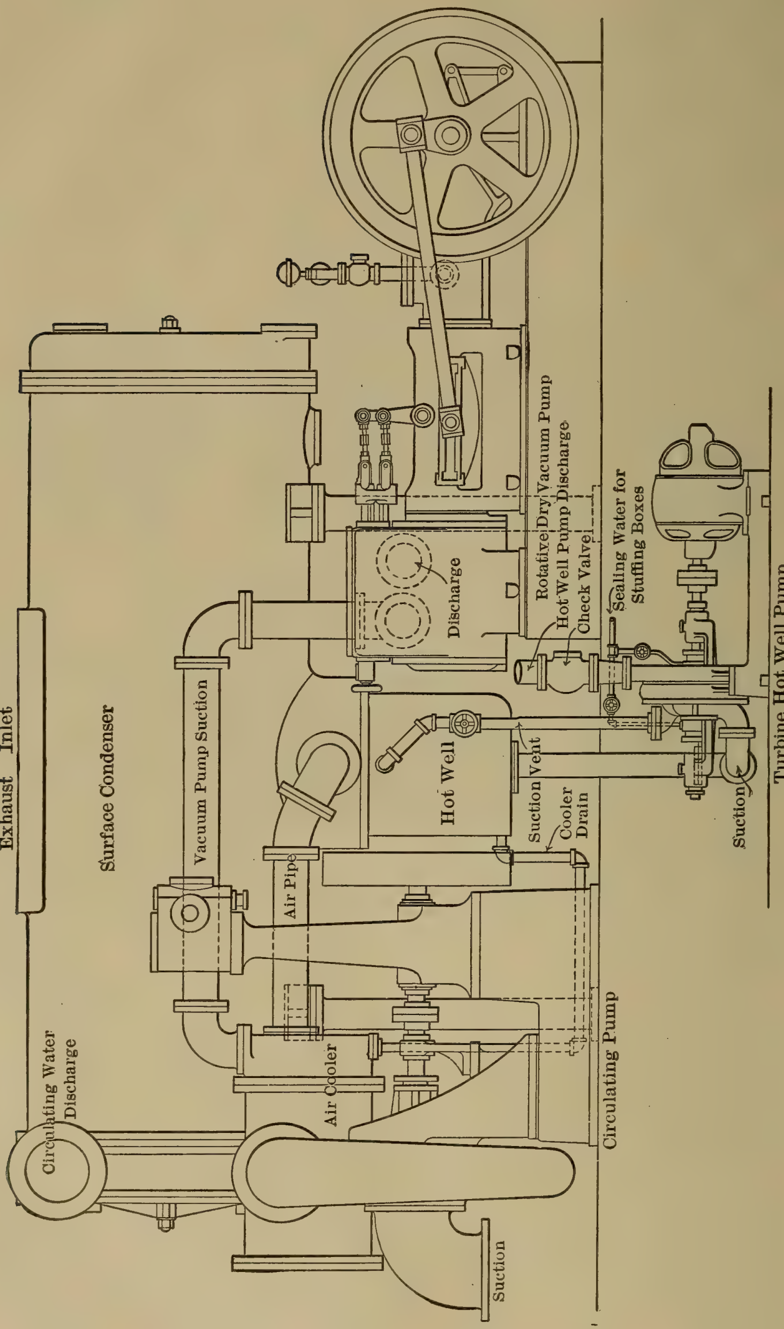


FIG. 312. Worthington High-vacuum System.

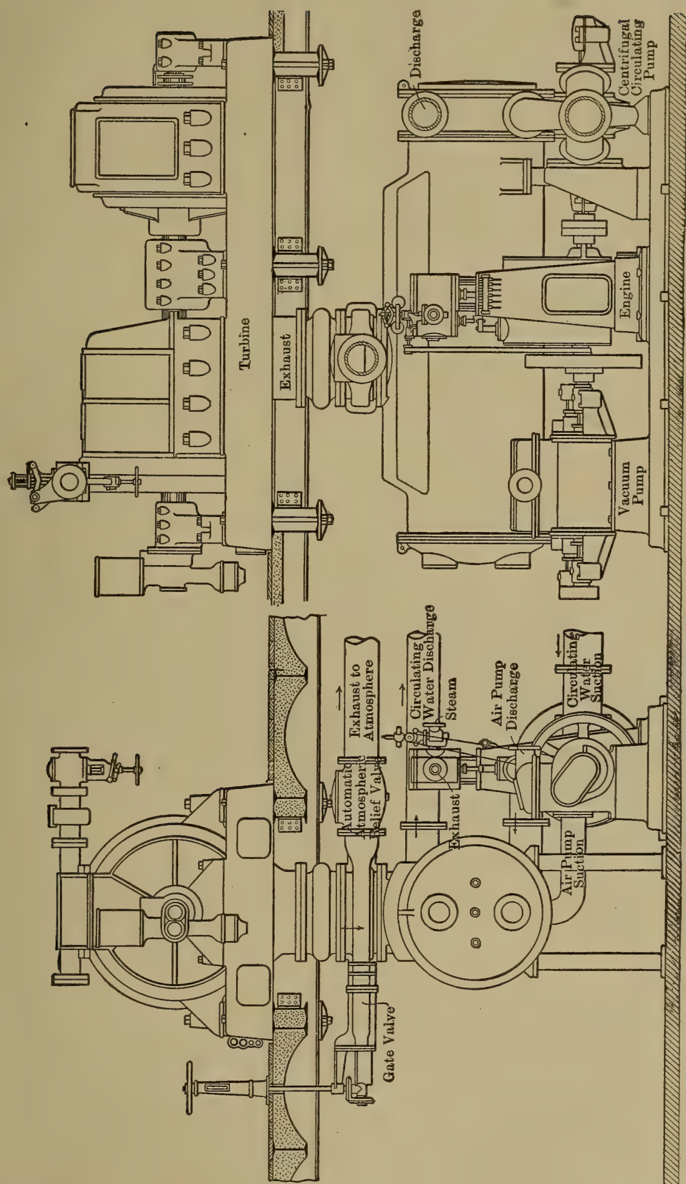


FIG. 313. C. H. Wheeler Company's High-vacuum System.

hot-well pump. The surface condenser is piped direct to the turbine exhaust, only a corrugated copper expansion joint and a tee intervening. A tubular water-vapor cooler, which is in reality a small surface condenser, is inserted in the circulating water line between the pump suction and condenser, and serves to arrest all the condensable vapor and thus reduces the volume to be handled by the air pump. All condensation, including that from the air cooler, collects in the hot well, from which it is pumped by a motor-driven circulating pump direct to heater or boiler. Cooling water is handled by a centrifugal pump having both suction and delivery pipes water-sealed, so that the work done by the pump is virtually that of overcoming the fluid friction in the condenser and piping. All valves and stuffing boxes are water-sealed

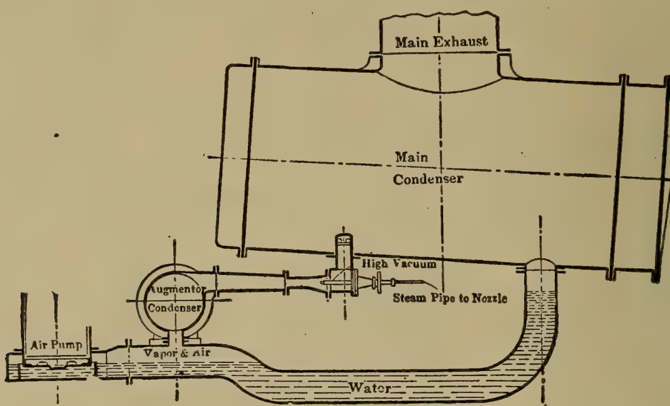


FIG. 314. Parsons Vacuum Augmenter.

to prevent any possible leakage of air, and the condenser pump cylinder is especially designed to avoid vapor binding. This makes it possible to maintain a vacuum of one-half pound absolute with cooling water at 60 degrees F. In the high-vacuum condenser installation of the Commonwealth Edison Company the dry-air pump and the circulating pump are direct connected to a single-cylinder Corliss engine.

Fig. 313 shows the general arrangement of the C. H. Wheeler Company's high-vacuum condensing outfit. The condensing chamber is shown in section in Fig. 310 and is described in paragraph 252. The wet-air pump is illustrated in Fig. 402 and is described in paragraph 324. No dry-air pump is needed, and the makers guarantee a vacuum within one inch of absolute under full-load conditions of steam turbine operation.

Fig. 314 shows a section through a Parsons "vacuum augmenter" for increasing the vacuum in a surface condenser. A pipe is led from

the bottom of the main condenser to an auxiliary or augmenter having about one-twentieth of the cooling surface of the main condenser. At the point indicated a small steam jet is provided which acts as an ejector and draws out the air and vapor from the condenser and delivers it to the air pump. The water seal prevents the air and vapor from returning to the condenser. With this arrangement, according to tests conducted by Mr. Parsons, if there is a vacuum of $27\frac{1}{2}$ or 28 inches in the

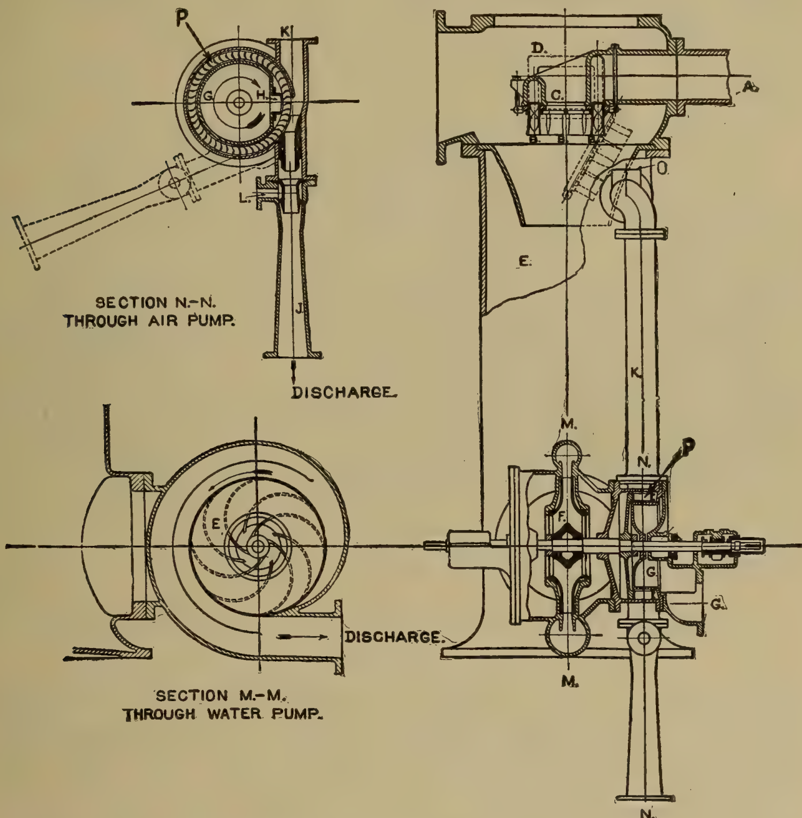


Fig. 315. Westinghouse-Leblanc Multi-jet High-vacuum Condenser System.

condenser, there may be only 26 at the air pump, which, therefore, may be of small size, the jet compressing the air and vapor from the condenser to about one-half of its original volume. The steam jet uses about one and one-half per cent of the steam used by the turbine at full load.

Jet Condensers. — Fig. 315 gives the general details of a Westinghouse-Leblanc multi-jet condenser which, under commercial conditions, has realized vacua within 99 per cent of the ideal. The most striking

feature of this system lies in its compactness and simplicity, a 1500-kilowatt equipment being less than 9 feet in height. Referring to Fig. 315, exhaust steam enters the condenser chamber at the upper left-hand opening and meets the cooling water as it is forced through spray nozzle C. The condensed steam and injection water fall to the bottom of the condenser and are removed by centrifugal pump M. The non-condensable vapors are withdrawn by valveless rotary air pumps P, through suction opening O. Referring to section N–N through the air pump it will be seen that this pump consists primarily of a reverse Pelton turbine wheel in conjunction with an ejector. Sealing water is introduced through the branch indicated by dotted outline, into the

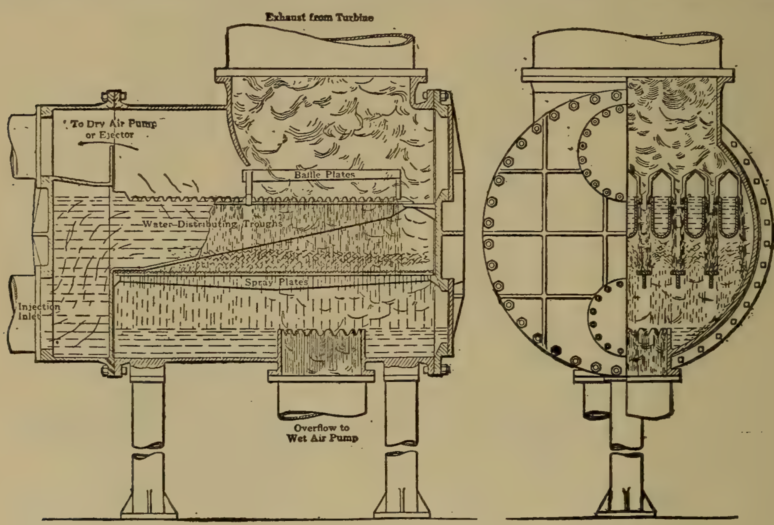


Fig. 316. Tomlinson Type C High-vacuum Jet Condenser.

central chamber G, from which it passes through port H. It is then caught up by the blades P of the Pelton wheel, which is rotated at a suitable speed, and ejected into the discharge cone in the form of thin sheets having a high velocity. These sheets of water meet the sides of the discharge cone and thus form a series of water pistons, each of which entraps a small pocket of air and forces it out against the atmospheric pressure. In passing through the air pump the sealing water receives practically no increase in temperature, hence the same water may be used over and over again. The air pump rotor and main pump runner are enclosed in a common casing mounted on the same shaft. This arrangement makes the plant very compact and requires the use of only one motor to drive both pumps. There is a clear passage through the condenser and pump, so that should the pump stop for any

reason air rushes into the condenser through the air pump and immediately breaks the vacuum. In starting up the condenser, steam is turned into auxiliary nozzle L, section N–N, for a few moments, thus creating sufficient vacuum to start the regular flow of water through the air pump. The pumps require from $1\frac{1}{2}$ to 3 per cent of the power generated by the main engines. Fig. 330 shows an application of a Westinghouse-Leblanc condenser to a Curtis turbine, and Fig. 331 the application of the Leblanc pumps to a surface condenser.

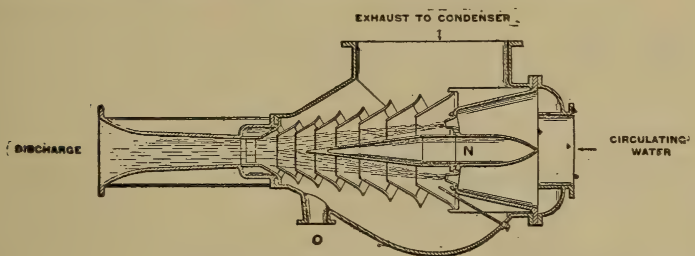


Fig. 317. Section through Condensing Chamber of Körting Multi-jet Condenser. Chamber Capable of Maintaining a Vacuum of 95 Per Cent of the Ideal without the use of Air Pumps.

254. Cooling Water, Surface Condensers. — The amount of cooling water required per pound of steam in a surface condenser is dependent upon the vacuum, the temperature of the condensed steam, and the range in temperature of the cooling water; thus:

$$W = \frac{H - t_1 + 32}{t_2 - t_0},^* \tag{173}$$

where

H = heat content of the exhaust steam above 32 degrees F.,
t_1 = temperature of the condensed steam,
t_0 = temperature of the injection water,
t_2 = temperature of the discharge water,
W = pounds of injection water necessary to condense one pound of steam.

Example: Required the quantity of cooling water necessary to condense one pound of steam under the following conditions: Initial temperature of the cooling water 60 degrees F.; final temperature 100 degrees F.; vacuum 26 inches, referred to 30-inch barometer. Here H = 1115 B.t.u., t_0 = 60, t_2 = 100.

$$W = \frac{1115 - 110 + 32}{100 - 60} = 25.9.$$

* See footnote, paragraph 243.

That is, the ratio of cooling water to condensed steam is approximately 26 to 1. In turbine practice where vacua as high as one-half pound absolute are obtained, the ratio of cooling water to condensed steam is nearly twice this quantity. For example, if a vacuum of 28.92 inches is desired with the barometer at 29.92 and the range of the circulating water temperature is 70 to 50 degrees and the temperature of the hot well 75 degrees, the ratio will be

$$W = \frac{1094 - 75 + 32}{70 - 50} = 52.3.$$

In determining the amount of cooling water it is well to bear in mind that in the ordinary condenser of the single- or double-flow type

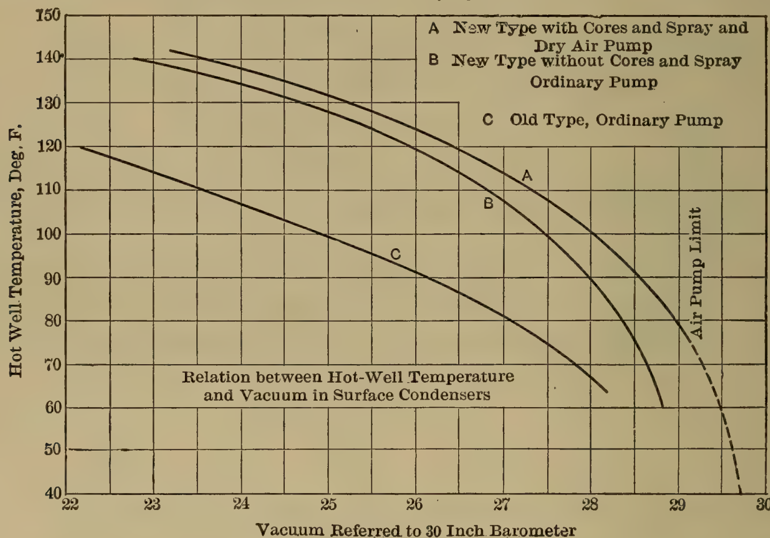


FIG. 318.

the temperature of the condensed steam will be from 10 to 20 degrees lower than that corresponding to the degree of vacuum in the condenser, and that the temperature of the condensing water at the discharge point will be from 5 to 10 degrees lower than the temperature due to the vacuum.

With well-designed condensers of the multi-flow type the temperature of the hot well may be from 0 to 5 degrees lower than the temperature due to the vacuum, and the temperature of the condensing water at the discharge point may be equal to that due to the vacuum. (Proc. Inst. of Naval Arch., March, 1906.) (See Fig. 318.)

255. Extent of Water-cooling Surface. — Theoretically, the operation of a surface condenser is divided into two periods, (1) the period

of condensation during which the heat of vaporization at the observed pressure is removed and (2) the period of cooling during which the temperature of the condensed steam is reduced. In order to determine accurately the extent of cooling surface it would be necessary to calculate the heat transmission for each of the two periods. In practice, however, it is assumed that condensation and cooling take place simultaneously, and that the mean temperature difference is a direct function of the temperature corresponding to the exhaust steam in the condenser and that of the condensed steam and cooling water. The error in these assumptions has only a slight influence on the estimation of the cooling surface and is entirely lost sight of in the liberal factor allowed in practice.

Let S = cooling surface in square feet,
H = heat content of the exhaust steam at condenser pressure,
t_0 = initial temperature of the circulating water,
t_2 = final temperature of the circulating water,
t_1 = final temperature of the condensed steam,
t_s = temperature of the exhaust steam at condenser pressure.
U = coefficient of heat transmission, B.t.u. per hour, per degree difference in temperature, per square foot of cooling surface,
d = mean difference temperature between t_s and t_2, and t_0,
W = weight of condensed steam per hour,

$$d = \frac{t_2 - t_0}{\log_e \dfrac{t_s - t_0}{t_s - t_2}} \text{ (see equation (210), Chapter XII);}$$

and since the heat absorbed by the cooling water is equal to the heat given up by the steam,

$$SUd = W\{H - (t_1 - 32)\}, \tag{174}$$

$$S = \frac{W(H - t_1 + 32)}{Ud}. \tag{175}$$

Whitham ("Steam-Engine Design," p. 283) uses the arithmetic mean $d' = t_s - \frac{t_0 + t_2}{2}$ instead of the mean as determined from Equation (210).

Equation (210) is based on the assumption that the fluid on each side of the tube is homogeneous, which is far from being true in the case of the air-steam mixture in a condenser, and for this reason many designers prefer to use the simpler arithmetic formula.

The coefficient of heat transfer, U, as used in above equations, refers to the *mean* or average value for the *entire* surface since the *actual* heat transmission varies widely for different parts of the condenser; thus

the actual value of U varies from over 1000 in the first few rows of the tubes (where the steam comes directly into contact with the cooling surface) to less than 50 in the bottom row (where the tubes are practically submerged in water of condensation) and to 3 or less for the tubes surrounded only by air.

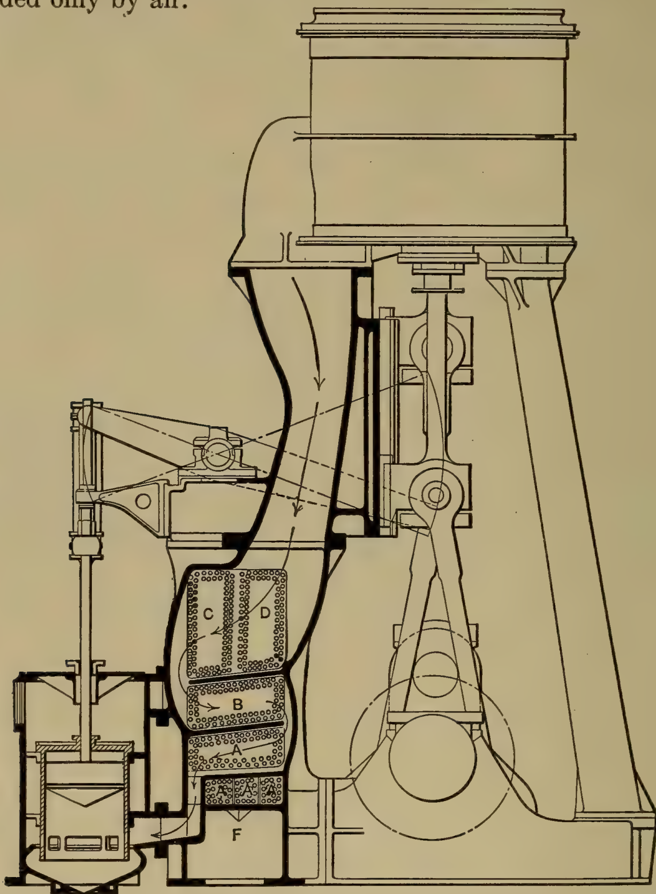


FIG. 319. Application of Weighton Dry-tube Surface Condenser to Vertical Marine Engine.

Professor Josse of the Royal Technical School, Charlottenburg, after an exhaustive investigation of the subject, found that the *actual* value of U varied with

(1) The material, thickness, shape, and cleanliness of the tube.
(2) The velocity of the water through the tubes.
(3) The velocity of the steam against the tubes.
(4) The percentage of air in the steam surrounding the tubes.
(5) The extent of submersion of the steam side of the tubes.

Some of the results of his investigations are shown in Figs. 320 to 322. See also Power and Engr., Feb. 2, 1909.

The effect of thickness, material, etc., of condenser tubes is so small in the ultimate result and the choice and arrangement are so largely determined by practical consideration that they may be neglected.

The value of U increases approximately as the square root of the velocity of the water flowing within the tube, so that *increase in water velocity effects a substantial increase in the heat transmission;* but the resistance encountered by the circulating water increases as the square of the velocity, and the power consumed in pumping the water increases as the third power of the velocity, so that a point is soon reached where the gain on the one hand may be offset by the loss on the other. See "The Transmission of Heat in Surface Condensation" by Geo. A. Orrok, Trans. A.S.M.E., vol. 32, p. 1138, 1910, for formulas pertaining to the value of U. This article contains, also, a complete bibliography on transmission of heat through tubes.

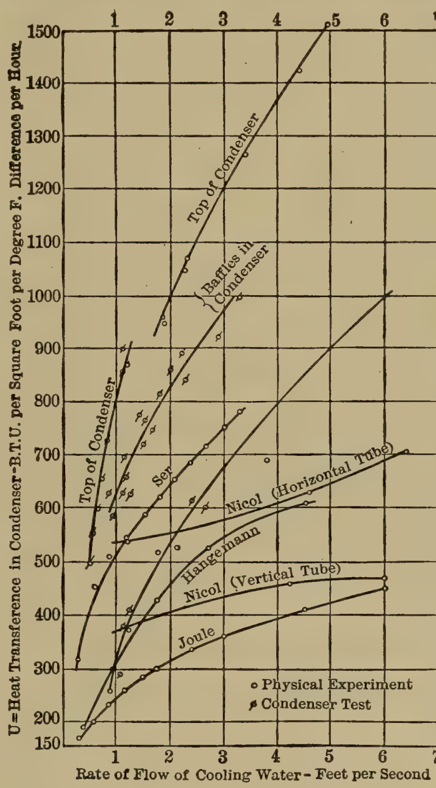


FIG. 320.

A study of a number of installations gave

Old-style surface condenser,

V = 30 to 240 feet per minute, average 90.

Modern dry-tube surface condenser,

V = 120 to 360 feet per minute, average 240.

From the curves in Figs. 321 and 322 it will be seen that air is an excellent heat-insulating material; hence, the greater the amount of air entrained with the steam the lower will be the coefficient of heat transmission. The necessity of removing the air as fast as it accumulates is at once apparent.

In the older types of surface condensers the water of condensation from the upper rows of tubes is permitted to fall on the rows immedi-

ately below, the water increasing in volume as it passes the successive banks of tubes until it completely envelops them. The coefficient U varies from 1000 or more in the upper row to less than 50 in the lower, giving a *mean* value of approximately 250 to 350 for the entire surface. In estimating the extent of cooling surface for a condenser of this type an average figure for plain brass tubes with water velocities of 50 to 100 feet per minute is $U = 250$. For a velocity of 100 to 240 feet per minute U may be taken 50 per cent greater than these figures. When the tubes are clean a much higher value may be taken, but a liberal factor is usually allowed for possible variation in the condition of operation.

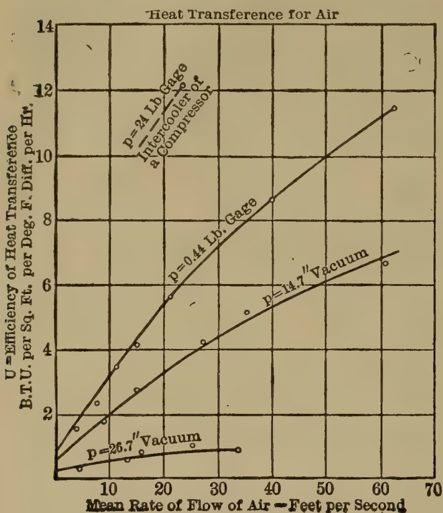


FIG. 321.

In the modern *dry-tube* surface condenser, designed along the lines of the one described in paragraph 253, in which the water of condensation is withdrawn as rapidly as it is formed and air entrainment is reduced to a minimum, *mean* values of U = 800 to 900 are not unusual. In estimating the extent of cooling surface for condensers of this type an average value of U is 600 with water velocities of 4 to 5 feet per second.

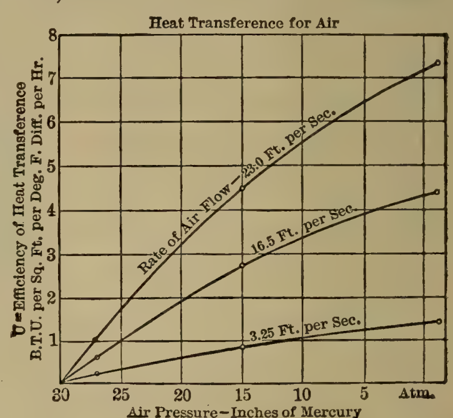


FIG. 322.

Example: Standard Type of Surface Condenser: — Required the number of square feet of cooling surface per i.h.p. necessary to condense the steam from an engine operating under the following conditions: Engine uses 20 pounds of steam per i.h.p. hour, vacuum 26 inches with barometer at 30; temperature of cooling water at 60 degrees.

Here $H = 1115$ and $t_s = 126$ (from steam tables),

$$t_0 = 60,$$

$$t_1 = t_s - 10 = 116.$$

In this type of condenser average practice gives a temperature difference of approximately 10 degrees between the temperature of the hot well and that corresponding to the degree of vacuum.

$$t_2 = t_s - 15 = 101.$$

Any value may be fixed upon for t_2 greater than t_0 and less than t_s. The nearer t_2 is to t_0 the greater must be the quantity of circulating water per unit of time for a given rate of condensation. On the other hand, the nearer t_2 is to t_s the less is the mean temperature difference d and hence the greater must be the cooling surface for a given coefficient of heat transmission. When water is cheap and the head pumped against is small t_2 should be given a lower value than when water is costly and the discharge head is large. Average engine practice, with conditions as stated, gives t_2 a value of approximately 15 degrees less than that corresponding to the degree of vacuum.

The logarithmic mean is, equation (210),

$$d = \frac{101 - 60}{\log_e \dfrac{126 - 60}{126 - 101}} = 42.4.$$

The arithmetic mean gives

$$d = 126 - \frac{60 + 101}{2} = 45.5.$$

Substitute the value of d in equation (175) and assume $U = 250$, the figure commonly used for this type of condenser.

$$S = \frac{20\,(1115 - 116 + 32)}{250 \times 42.4} = 1.94,$$

or say two square feet per i.h.p. of engine.

Surface condensers of this type are ordinarily rated on a basis of two square feet per i.h.p.

Example: Dry-tube Multi-flow Surface Condenser: — Required the number of square feet of cooling surface per kilowatt necessary to condense the steam from a steam turbine operating under the following conditions: Turbine uses 15 pounds of steam per kilowatt-hour; vacuum 28.5 inches, referred to 30-inch barometer; temperature of cooling water 70 degrees.

Here
$$H = 0.9 \times 1100 = 990.$$

The total heat of dry steam corresponding to an absolute pressure of 1.5 inches is 1100, but in the case of high-vacuum turbine practice the steam entering the condenser is far from being dry, the quality varying from 0.80 to 0.95, depending upon the quality of the steam at admission. An average correction factor is 0.9.

$$t_s = 92, \quad t_0 = 70, \quad t_1 = t_s - 4 = 88.$$

In this type of condenser the hot-well temperature varies from $t_1 = t_s$ to $t_1 = t_s - 8$. $t_2 = t_s - 5 = 87$.

In this type t_2 varies from $t_2 = t_s$ to $t_2 = t_s - 10$.

$$d = \frac{87 - 70}{\log \frac{92 - 70}{92 - 87}} = 11.5.$$

Arithmetic mean gives $d = 92 - \frac{70 + 87}{2} = 13.5$.

Substitute the value of d in equation (210) and assume $U = 600$, the figure commonly used for this type of condenser.

$$S = \frac{15\ (990 - 88 + 32)}{600 \times 11.5} = 2.02,$$

or say 2 square feet per kilowatt of generator. There is no standard rating of surface condenser for steam-turbine work because of the wide variation in operating conditions. A study of a number of modern installations gives

1.6 to 2.5 square feet per kilowatt for large turbo-generators using dry-tube surface condensers.

2.5 to 4 square feet per kilowatt for small turbo-generators using standard surface condensers.

Professor Weighton found from his experiments that a surface condenser constructed on the lines of the one described in paragraph 253 in conjunction with dry-air pumps, was capable of condensing 20 pounds of steam per square foot of surface per hour and maintained a vacuum of $28\frac{1}{2}$ inches (referred to a 30-inch barometer), and this with a cooling-water consumption of 24 pounds per pound of condensed steam; with an inlet temperature of 50 degrees F. a condensation of 35 pounds of steam per hour per square foot of cooling surface was effected at a ratio of 28 pounds of cooling water per pound of steam, the vacuum remaining $28\frac{1}{2}$ inches. See Fig. 318. (Engineering Record, May 19, 1906, p. 615.)

TABLE 81.

EXAMPLES OF MODERN CONDENSER PROPORTIONS.

Name of Station.	Size of Turbo-Generators.	Sq. Ft. of Condenser Surface.	Sq. Ft. of Surface per Kw.
Commonwealth Edison Co.:			
Northwest Station	20,000	32,000	1.60
Quarry Street	14,000	25,000	1.79
Fisk Street	12,000	25,000	2.08
*59th St., Interborough, N. Y.	15,000	25,000	1.67
Metropolitan St. Ry., Kansas City	10,00	22,000	2.20

* Combined Engine and Low-pressure Turbine.

The curves in Fig. 323 are based upon equation (175) with $U = 300$ and afford a simple means for determining the extent of cooling surface for different conditions of operation. For any other value of U multiply by 300 and divide by the new value of U.

Surface Condenser Air Pumps. — See paragraphs 319 to 328.

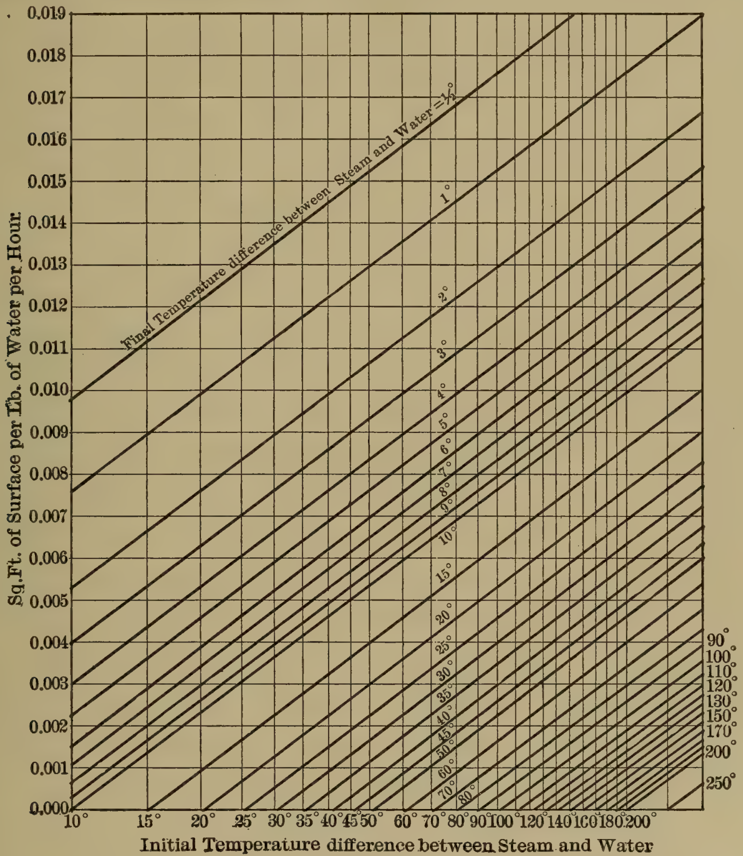


Fig. 323. Curves for Determining the Amount of Cooling Surface.

256. Dry-air Surface Condensers (Forced Circulation). — Where water is very scarce and the feed supply is reclaimed by condensing the exhaust steam, water-cooled condensers may be prohibitive in cost of operation, even when combined with cooling tower or other water-cooling device, since the latter involves a loss of water approximately equivalent to the amount of steam condensed, due to evaporation.

Under these conditions air cooling has been successfully adopted. In the city of Kalgoorlie, West Australia, an electric station of 2000-horse-power capacity is equipped with air-cooled surface condensers.

The condensers have been in use five years (1906), and have given excellent service with very little expense and maintenance. The condenser consists of a large number of narrow chambers constructed of thin corrugated sheet-steel plates spaced $\frac{1}{4}$ inch between centers. Each chamber has 1345 square inches of cooling surface. Fifty-one of these chambers are grouped into a compartment and 15 compartments constitute a section. Each section is equipped with three motor-driven fans 7 feet in diameter and running normally at 320 r.p.m. In all there are six sections, giving a total cooling surface of 45,000 square feet. The steam consumption of the main engines is 16 to 16.5 pounds per i.h.p. hour at rated load. At full load the fans require 130 kilowatts, or approximately 10 per cent of the station output. The average vacuum obtained is about 18 inches throughout the year and ranges from 0 inches on very hot days to 22 inches in cooler weather. The following figures, based on actual observation, show the effect of temperature of the external air on the vacuum when condensing 32,000 pounds of steam per hour (the rated capacity of the condenser).

Temperature External Air, Degrees F.	Vacuum, Inches (referred to 30-Inch Barometer).	Temperature External Air, Degrees F.	Vacuum, Inches (referred to 30-Inch Barometer).
42.8	22	96.8	9.6
50	21.2	100.4	7.6
60.8	20	107.6	3.6
68	18.4	113	0
78.8	16		

Air-Cooled Surface Condensers: Engineering News, Oct., 1902, p. 271; ibid., Vol. 49, p. 203.

257. Quantity of Air for Cooling (Dry-air Condenser). — The volume of air, under atmospheric conditions, necessary to condense steam to any given temperature may be determined as follows:

Let H = heat content of the steam at condenser pressure,
t_s = temperature of the vapor in the condenser,
t_1 = temperature of the condensed steam,
t = temperature of the air entering condenser,
t_0 = temperature of the air leaving condenser,
V = volume of air in cubic feet necessary to condense and cool one pound of steam,
B = specific weight of air under atmospheric conditions,
C = mean specific heat of air under atmospheric conditions,
d = mean temperature difference between the air and steam,
S = cooling surface in square feet,
U = coefficient of heat transmission, B.t.u. per square foot per degree difference in temperature per hour.

Since the heat absorbed by the air must be equal to the heat given up by the steam, neglecting radiation, we have

$$VBC\ (t_0 - t) = H - t_1 + 32, \tag{176}$$

from which

$$V = \frac{H - t_1 + 32}{BC\ (t_0 - t)}. \tag{177}$$

For practical purposes C may be taken as the specific heat of dry air, the error due to this assumption being negligible even if the air is saturated with moisture.

Example: How many cubic feet of air are necessary to condense and cool one pound of steam under the following conditions: Vacuum 20 inches; temperature of entering air, leaving air, and condensed steam, 60, 110, and 140 degrees F. respectively?

Here $H = 1130$ (from steam tables),
$t_0 = 110$, $t_1 = 140$, $t = 60$, $C = 0.24$, $B = 0.075$.

Substituting these values in equation (177),

$$V = \frac{1130 - 140 + 32}{0.075 \times 0.24\ (110 - 60)}$$ = 1135 cubic feet of air necessary to condense one pound of steam under the given conditions.

The proper area of cooling surface depends upon the value of the coefficient of heat transmission, which varies with the velocity and humidity of the air and character of the cooling surface. Accurate data are not available on this point.

A few experiments made at the Armour Institute of Technology gave values of U = 10 to 25 B.t.u. per hour, per square foot, per degree difference in temperature for air velocities of 500 to 4000 feet per minute for corrugated-steel sheeting $\frac{1}{8}$ inch thick. Hence, substituting in equations (175) and (177) we get, for the above example, $S = 1.5$ square feet of cooling surface per pound of steam condensed per hour for air velocity of 4000 feet per minute, and $S = 3.7$ square feet for a velocity of 500 feet per minute.

258. Saturated-air Surface Condensers (Natural Draft).—Fig. 324 shows vertical and horizontal sections of a Pennel saturated-air surface condenser. The apparatus consists of an upright cylindrical shell containing a number of vertical 4-inch steel tubes through which air is drawn by natural draft. A centrifugal pump circulates about one half gallon of water per horse power per minute from a cistern below the condenser. The water flowing over the upper tube sheet and then descending the tubes by gravity forms a film over their entire interior surface.

The condensing action is as follows: The current of exhaust steam entering the side of the shell at A is caused by suitable baffle plates to

circulate among the tubes, and in condensing gives up its latent heat to the water film, which wholly or partially evaporates, saturating the ascending current of air at its own temperature. The upward current of hot vapor-laden air carries off the heat into the atmosphere. The cooling water which is not evaporated and lost to the atmosphere falls into the cistern below to be again taken up by the circulating pump, the water level in the cistern being kept constant by a float governing a valve on the supply pipe. The non-condensable gases collect at *C*,

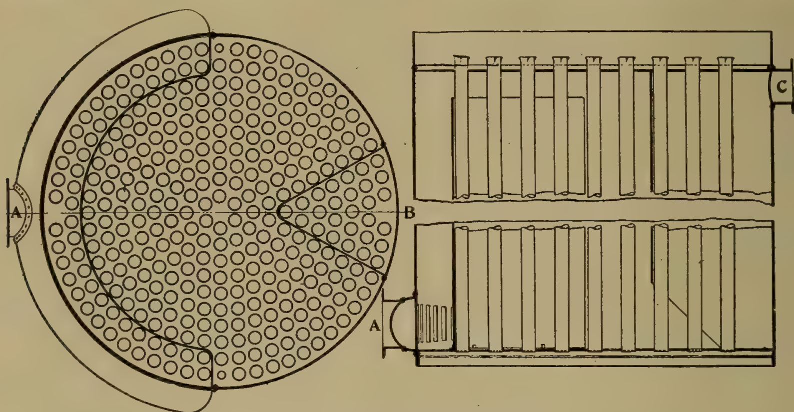


FIG. 324. Pennel Saturated-air Surface Condenser.

where they are removed by the dry-air pump, while the condensed steam is drawn off from the bottom tube sheet by the vacuum pump and discharged into the hot well. An excellent feature of this device is that the film of water on the cooling surface is secured without interference with the ascending air currents and also without the use of sprays through small orifices likely to become clogged with rust or sediment. Where the recovery of the condensed steam is essential and a high vacuum of secondary importance, condensers of this type have proved to be good investments on account of the low first cost.

Table 82 gives the results of a test of a condenser of this type, taking steam from a 30 × 58 × 48 engine running at 45 r.p.m. (Power, December, 1903, p. 672; West. Elect., May 19, 1900, p. 323.)

TABLE 82.

TEST OF PENNEL SATURATED-AIR SURFACE CONDENSER.

Duration of trial	9 hours
Average steam pressure at engine by gauge	139.8 pounds
Average vacuum, mercury column	17.5 inches
Average temperature in condenser	123.7 degrees F.
Average temperature of circulating water	116.4 degrees F.

TABLE 82 (*Continued*).

Average temperature of city water	52 degrees F.
Average temperature of outside air	62 degrees F.
Average temperature of saturated air	106 degrees F.
Average draft in stack of condenser	1.1 inches
Average humidity of outside air	67 per cent
Average amount of steam condensed per hour	7950 pounds
Average amount of circulating water used per hour	114,660 pounds
Average amount of city water used per hour	3462 pounds
Pounds of city water per pound of steam	2.3
Pounds of circulating water per pound of steam	14.4
Average horse power of engine	569.7
Steam, pounds per i.h.p. per hour	13.95
Horse power required to run air pumps	10.5
Horse power required to run circulating pumps	3.0
Condensing surface, square feet	3900
Pounds of steam condensed per square foot surface per hour	2038
Barometer	28.58 inches
Vapor tension corresponding to 123.7 degrees	3.82 inches
Per cent of main engine steam used by auxiliaries	2.38

Fig. 325 illustrates the Pennel "flask" type of atmospheric condenser. The exhaust steam enters below and follows the zigzag course bounded by the internal stay channels, condensing as it goes and driving before it the non-condensable gases to the outlet at the top. The condensed steam gravitates to the bottom and thence to the hot well. The top of the flask is trough shaped and causes the cooling water to flow down the sides of the flask in a thin stream. The portion of the cooling water not evaporated collects at the bottom of the flask and flows to the cooling-water reservoir.

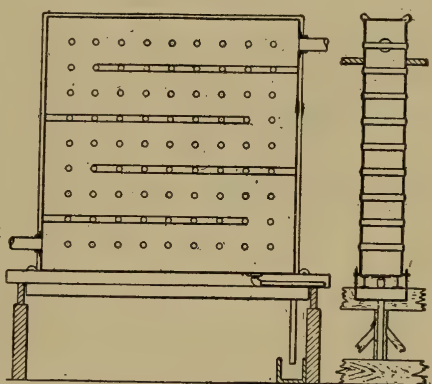


Fig. 325. Pennel Flask Type of Saturated-air Surface Condenser.

259. Evaporative Surface Condensers. — An evaporative surface condenser consists of a number of copper, brass, wrought- or cast-iron tubes arranged horizontally or vertically and connected to manifolds or chambers at each end. The exhaust steam passes through the tubes and a thin film of water is allowed to flow over the external surfaces. The cooling effect is brought about by the evaporation of part of the circulating water, and the general principle of operation is the same as that of the saturated-air condenser described above. Evaporation is sometimes hastened by constructing a flue over the

tubes, thereby creating a natural draft, or by means of fans. With horizontal cast-iron tubes and natural draft, vacua from 23 to 27 inches are readily maintained with a cooling surface of approximately eight tenths square foot per pound of steam condensed per hour. With vertical brass tubes and fan draft 8 pounds of steam per hour per square foot of cooling surface is not an unusual figure. The amount of cooling water evaporated per pound of steam varies from eight tenths to one pound, depending upon the draft. The power necessary to operate the pumps and fans varies from 1 to 10 per cent of the total output of the plant. For an interesting discussion of evaporative condensers the reader is referred to the admirable article by Oldham in the Proceedings of the Institute of Mechanical Engineers, 1899, and reproduced as a serial in Engineering (London), April 28 to June 30, 1899. The following test of a vertical cast-iron tube evaporative surface condenser (Table 83) will give some idea of the performance of this type of condenser. This condenser consisted of two rows of 4-inch vertical cast-iron pipes connected at the top by U bends and at the bottom by cast-iron manifolds. A perforated iron trough distributes the water over the center of the bend and causes it to flow in a thin stream over the surface of the tubes. A wet-air pump is used for withdrawing the condensed steam and air. No fan is used for hastening evaporation.*

TABLE 83.

TEST OF A CAST-IRON, VERTICAL-TUBE, EVAPORATIVE SURFACE CONDENSER NATURAL DRAFT.

Date	Sept. 12	Sept. 13
Weather	Wet	Fine
Barometer	29.8	29.5
Temperature of air	?	60
Cooling surface, external	272	272
Duration of trial, minutes	99	115
Weight of steam condensed, pounds	800	800
Boiler pressure	60	60
Weight of water in circulation	1830	1830
Weight of fresh water added	600	640
Vacuum in condenser	23.36	24.1
Initial temperature of circulating water	117.5	113.9
Final temperature of circulating water	128.4	125
Temperature of "make up" water	58	58
Temperature of water in hot well	136.5	131.8
Weight of steam condensed per hour, pounds	485	427
Weight of water circulated per hour, pounds	6786	?
Weight of "make-up" water added per hour	364	334
Weight of steam condensed per square foot of cooling surface per hour	1.8	1.54
Weight of "make-up" water per pound of steam condensed, pounds	0.75	0.80

* See end of paragraph 27 for evaporative surface condenser calculations.

Evaporative Condensers: Engr., Lond., May 5, 1899, pp. 432, 442, 447; Engineering, May 19, 1899, p. 661, June 2, 1899, p. 721, June 30, 1899, p. 861; Trans. A.S.M.E., 14-696; Power, Nov. 16, 1909; Prac. Engr. U. S., June, 1910, p. 346.

260. Location and Arrangement of Condensers. — In the modern power house one sees two general arrangements of condensers and auxiliaries:

1. The independent or subdivided system, in which each engine or turbine is provided with its own condenser, air and circulating pumps.
2. The central system, in which the condensers and auxiliaries are grouped together. Ordinarily one condenser suffices for all engines.

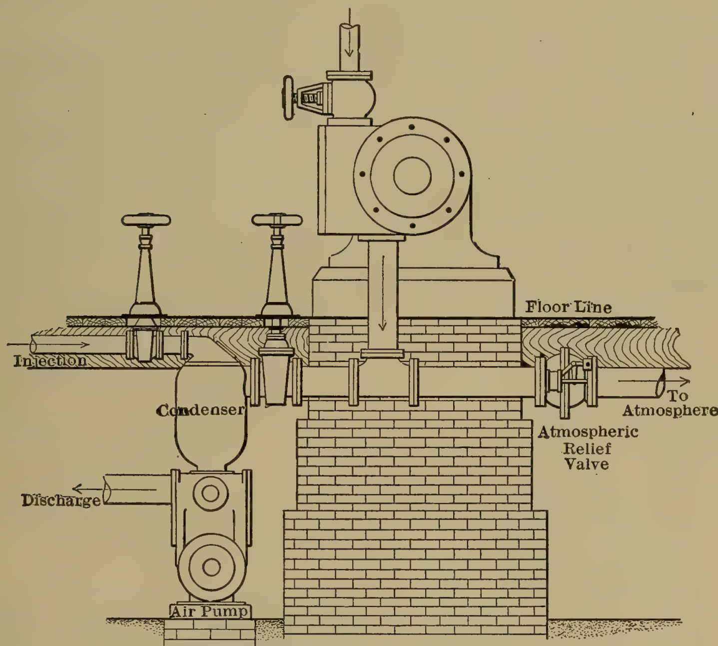


FIG. 326. Jet Condenser Located Below Engine-room Floor.

THE INDEPENDENT SYSTEM. — The condenser is usually placed close to and below the engine so that all condensation may gravitate into it. Figs. 326 and 329 show an application of this system with jet condensers. Here each condenser receives its supply of cooling water from a main injection pipe and discharges into a main overflow pipe. The exhaust pipe leading to the condenser is by-passed through a suitable atmospheric relief valve to a main free exhaust header so that the engine may operate non-condensing in case the vacuum breaks or the condenser is cut out. The chief feature of this arrangement is its flexibility, as each unit is complete in itself and independent of the others.

By far the greater number of central stations are equipped with independent condensers.

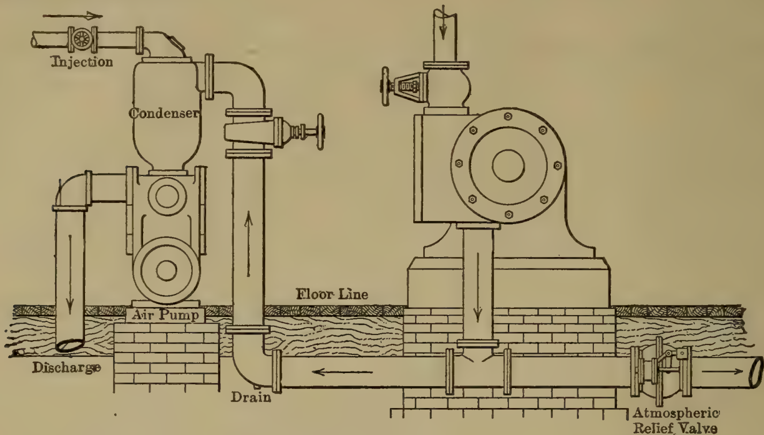


FIG. 327. Jet Condenser Located Above Engine-room Floor.

Occasionally a jet condenser is located on the same level with the engine or even above it, Fig. 327, but such a location should be avoided if possible, as it usually necessitates a larger number of bends and

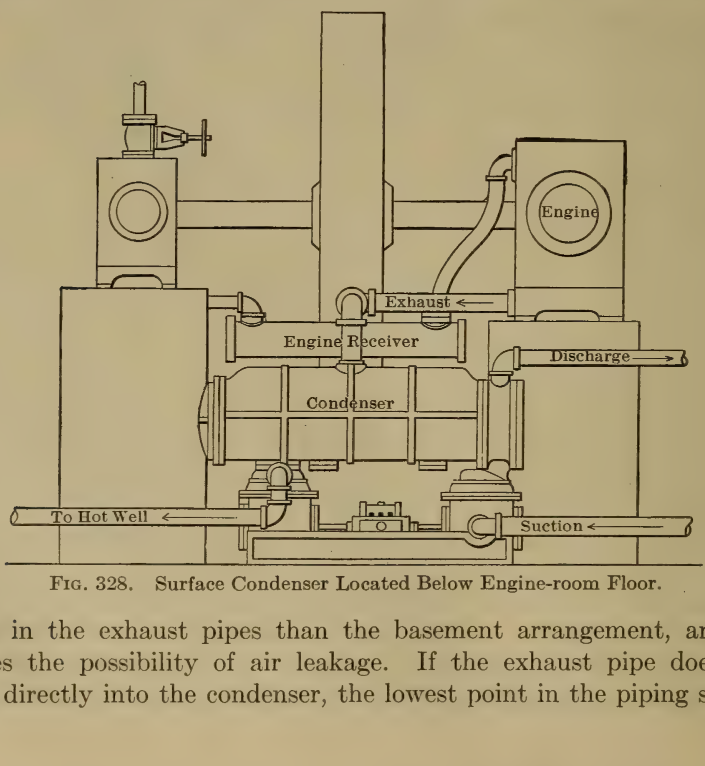


FIG. 328. Surface Condenser Located Below Engine-room Floor.

joints in the exhaust pipes than the basement arrangement, and increases the possibility of air leakage. If the exhaust pipe does not drain directly into the condenser, the lowest point in the piping should

always be provided with a drip which should be opened when the engine is shut down, as condensation and leakage are apt to fill the pipe with water if the engine stands for any length of time. The end of the drip should be connected so that water cannot be drawn back through the drip pipe and into the engine cylinder. The length of exhaust pipe and particularly the number of bends between engine and condenser should be kept as small as possible, otherwise the engine may not derive the full benefit of the vacuum in the condenser. A case is recorded where the exhaust piping and appurtenances in connection with a 5000-horse-power engine caused a drop of several inches in vacuum between condenser and exhaust opening of the low-pressure cylinder. (National Engineer, December, 1906, p. 10.) The wet-air pump must always be located below the condenser chamber so that the condensation may gravitate to it.

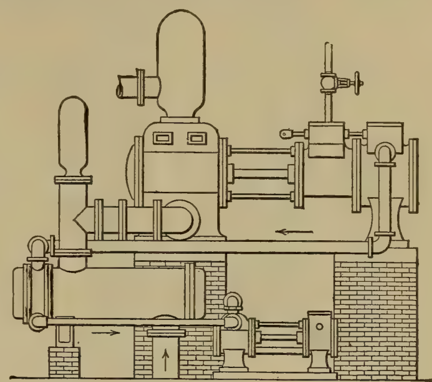


Fig. 329. Surface Condenser Installed in Connection with Pumping.

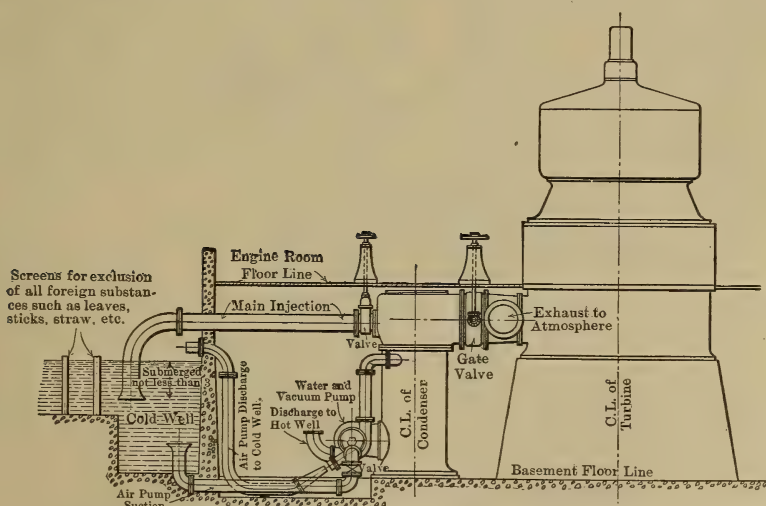


Fig. 330. Westinghouse-Leblanc Condenser Installation Engine.

Fig. 328 shows the arrangement of a surface condenser with combined air and circulating pump in connection with a horizontal cross compound engine. The condenser and appurtenances are placed below

the engine, thereby permitting the condenser to be closely connected to the engine.

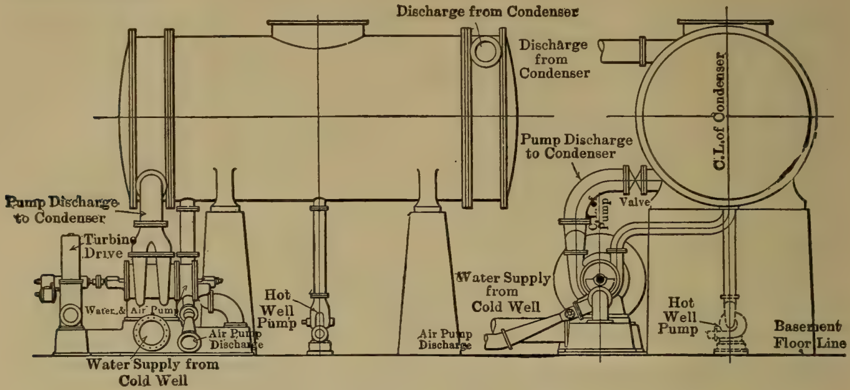


FIG. 331. Surface Condenser with Leblanc Pumps.

Fig. 329 shows the arrangement of a surface condenser in connection with a pumping engine. The condenser is placed in series with the pump suction.

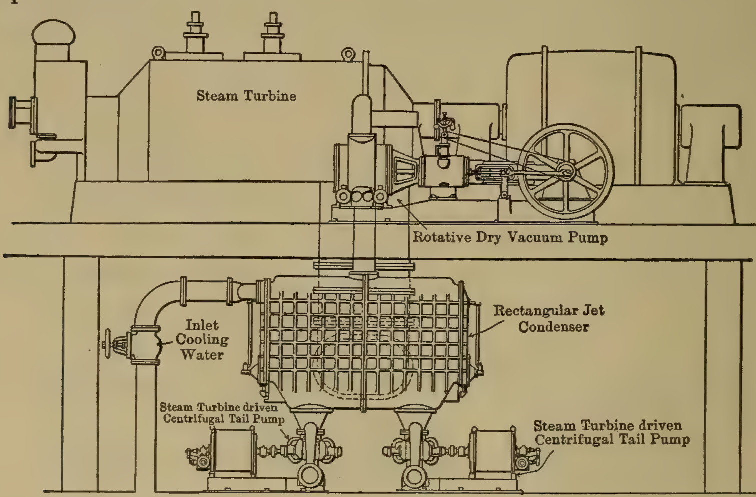


FIG. 332. Wheeler Rectangular Jet Condenser with Centrifugal Tail Pump and Rotative Dry Vacuum Pump in Connection with a 10,000-kilowatt Steam Turbine.

CENTRAL SYSTEMS. — In the central condensing systems the condenser is located at any convenient point and the exhaust from all the engines piped to it. Any arrangement of condenser and auxiliary machinery may be adopted which will favor the lowest cost of installation and expense of operation. Except where continuity of operation is absolutely essential, only one circulating pump and one air pump

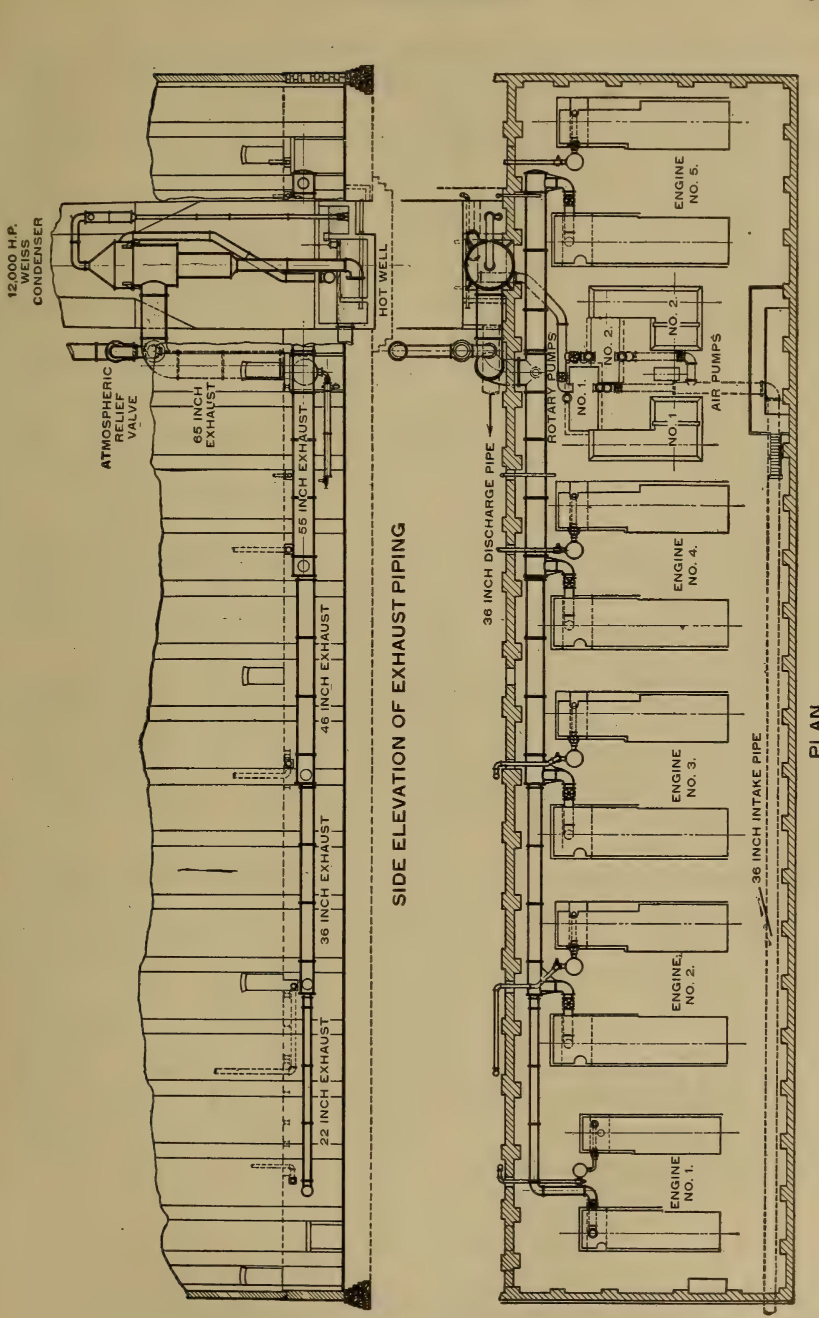


FIG. 333. Condenser and Exhaust Steam Piping at the Northwestern Elevated R.R. Power House, Chicago, Ill.

are installed. This reduces the number of auxiliary pumps and appliances to a minimum, with a consequent decrease in first cost and maintenance. With properly designed exhaust piping the condenser may be located at a considerable distance from the engine without undue loss of vacuum.

Central condensers have found great favor in power plants in which the individual units are subjected to extreme variations in load, as in rolling mills. At the works of the Illinois Steel Company, South Chicago, Ill., one condenser takes care of the steam from 15,000 horse power of engines in the rail mill, and another condenses the steam from the 15,000 horse power of engines in the Bessemer steel mill. A notable installation of this system in connection with street-railway work is in the power house of the Northwestern Elevated Company, Chicago, where a single condenser takes care of the exhaust steam of five engines, 11,000 horse power in all. Fig. 333 shows the general arrangement of this installation.

For a comparison of the advantages and disadvantages of the independent and central systems see Engineering Magazine, October, 1900, p. 56, Engineering, London, June 23, 1899, p. 615, and Engineering, July 17, 1903.

261. Power Consumption of Condenser Auxiliaries. — In estimating the cost of producing vacua with the different types of auxiliaries, steam driven, electrically driven, or belted, the power consumption is most conveniently expressed in terms of the equivalent *heat consumption* of the auxiliary in question and not the indicated or developed power. For example, suppose a power plant has a number of 1200-i.h.p. engines direct connected to 800-kilowatt generators and that the engines use 20 pounds of steam per i.h.p. hour at rated load; furthermore suppose the engine driving the air pump (jet condenser) to indicate 24 horse power. Now, it is manifestly incorrect to say that the power consumption of the air pump is equivalent to $\frac{24}{1200} = 2$ per cent of the main engine power unless the engine driving the air pump uses 20 pounds of steam per i.h.p. hour. As a matter of fact the small engine probably uses 30 to 40 pounds or more of steam per i.h.p. hour, and the true power consumption is

$$\frac{24 \times 30}{1200 \times 20} = 3 \text{ per cent, or more.}$$

If the exhaust steam is piped to the condenser, then all of this 3 per cent or more should be charged against the condenser; if the steam is piped to a heater, then only the difference between the heat entering

the small engine and that given up to the feed water should be charged against it. For example, suppose the engine in the preceding examples uses 30 pounds of steam per i.h.p. hour when running condensing and 40 pounds when operating non-condensing. Let the initial steam pressure be 150 pounds and feed-water temperature 120 degrees F. when the air pump is running condensing. If the boiler feed is not taken from the hot well, the heat in the exhaust steam is lost so far as the economy of the plant is concerned, and the heat consumption per i.h.p. hour is $30 \{1193.6 - (120 - 32)\} = 33{,}168$ B.t.u. This represents the cost, in heat units, of producing the vacuum, and is equivalent to 3 per cent of the main engine output.

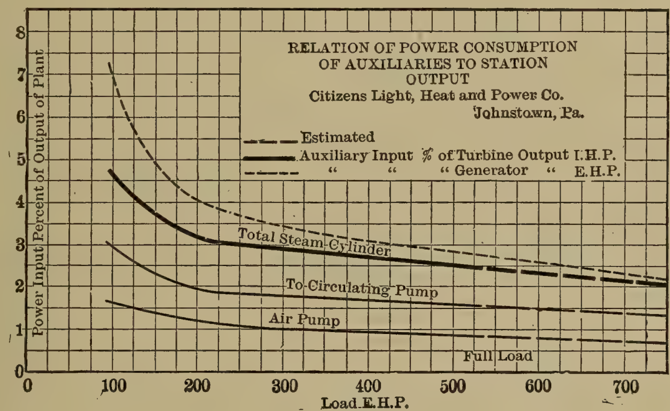


FIG. 334.

If the air pump runs non-condensing and the exhaust steam is piped to the heater, each pound of exhaust steam gives up approximately 950 B.t.u. per hour to the feed water and the temperature of the latter is raised from 120 to 180 degrees F. The heat entering the air pump is $40\{1193.6 - (120 - 32)\} = 44{,}224$ B.t.u. per i.h.p. hour. But $40 \times 950 = 38{,}000$ B.t.u. are returned to the feed water. Hence $44{,}224 - 38{,}000 = 6224$ is the net heat consumption of the air pumps per i.h.p. hour. This corresponds to approximately 0.55 per cent of the main engine output.

In the preceding example suppose the air pump to be motor driven and that it requires 20 electrical horse power per hour. This will be the equivalent of $\frac{20}{0.85 \times 0.90} = 26.2$ i.h.p. of the main engine on the assumption that the efficiency of the small motor is 85 per cent and that of the engine and generator combined 90 per cent. The power required by the air pump will be $26.2 \div 1200 = 2.2$ per cent of the total output.

In practice the auxiliaries use the equivalent of from 1 to 15 per cent of the main engine or turbine steam, depending upon the size of the plant, character and number of auxiliaries, and the conditions of operation.

Table 84 gives the power consumption of the condenser auxiliaries in a number of installations. Fig. 334 shows the relation between the power consumption of the auxiliaries and the total output of the station at different loads for a Parsons steam-turbine installation, and Fig. 334 shows a similar relation for a 2000-kilowatt Curtis turbine. (J. R. Bibbins, Power, January, 1905.)

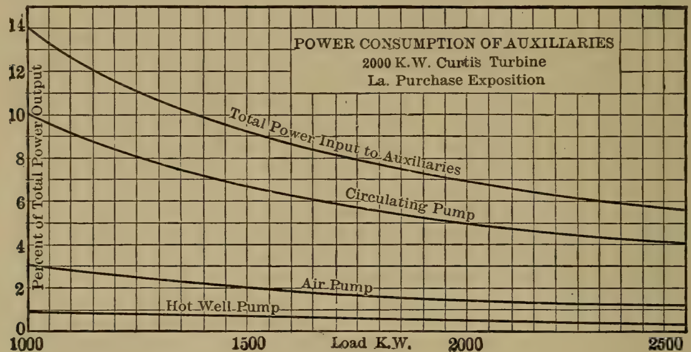


Fig. 335.

262. Cost of Condensers. — The following figures give an idea of the relative costs of the different types of condensers and auxiliaries for a 1000-i.h.p. plant using 20 pounds of steam per i.h.p. hour at rated load, or a total of 20,000 pounds per hour. Vacuum to be maintained, 26 inches, unless otherwise stated; temperature of cooling water, 70 degrees F.; hot-well temperature, 105 to 120 degrees F.; distance between engine exhaust opening and mean level of intake well, 10 feet.

Siphon Condensers.

1 16″ siphon condenser with 6″ centrifugal pump driven by 6″ by 6″ vertical engine $800

Jet Condensers.

1 14″ by 22″ by 24″ jet condenser with single horizontal direct-acting pump 1335

1 16″ by 24″ by 18″ jet condenser with single vertical direct-acting pump 1620

1 14″ by 24″ by 18″ jet condenser with single vertical flywheel vacuum pump 1770

1 12″ by 17″ by 22″ by 25″ jet condenser, single horizontal direct-acting compound pump 2200

TABLE 84.

POWER CONSUMPTION OF CONDENSER AUXILIARIES.

Type of Condenser.	Method of Operating Pumps.	Cooling Water, Deg. F.	Vacuum, Inches.	Barometer.	Absolute Pressure, Inches.	Steam Condensed, Lbs. per Hour.	Ratio of Cooling Water to Steam Condensed.	Power Consumed by Pumps.	Per Cent of Total Power Used by Pumps.	Head Pumped Against, F.	References.
Hulton, ejector	Electrical.	67.6	26.17	30.05	3.88	6,966	68.7	22.5 Kw.	7.1	47.5	Proc. Inst. of Electrical Engineers, Jan. 19, 1905.
Neepsend, surface	do	65.3	27.7	29.9	2.2	29,530	30	37.4 Kw.	2.3	20	
Bromwich, surface	do	51	28.2	29.8	1.6	10,000	50	9.17 Kw.	2.05	15	
Manchester, jet	do	82	26.6	29.5	2.9	35,700	......	40 Kw.	2.2	40	Boston, Edison Co.
Surface	....	66.9	28.13	29.93	1.8	95,200	90	102.7 I.h.p.	2.5	......	
Weiss, barometric	Steam	50	27	30	3	128,000	18	80 I.h.p.	1.1	46	N. W. El., Chicago.
Alberger, surface	do	65	28	30.03	2.03	105,000	81.8	120 I.h.p.	1.5	......	Fisk Street Station, Chicago, Edison.
Surface	do	......	......	......	......	10,250	......	....	8.1	......	Nashua Light, Heat, and Power Co.
Tomlinson, barometric	do	40	27.8	30	2.2	70,000	30	45 I.h.p.	1	48	South Side Alley El., Chicago.
Surface	Electrical	60	28	29.5	1.5	32,000	......	83 Kw.	4.1	30	Los Angeles.
Leblanc, jet	Steam	65	28.8	30.0	1.2	41,100	82.5	52 H.p.	......	......	Shop test.

Barometric Condensers.

1 barometric condenser, 10″ by 16″ by 12″ horizontal single-cylinder rotative dry-air pump; 8″ horizontal volute centrifugal pump direct connected to 23-horse-power high-speed engine	$2500
1 barometric condenser, 16″ by 16″ dry-air pump direct connected to 9″ by 16″ steam engine; positive rotary pump, for circulating cooling water, belted to above engine	4300

Surface Condensers.

1 surface condenser, 1025 square feet cooling surface, mounted over 7½″ by 14″ by 14″ by 12″ combined air and circulating pump	2100
1 surface condenser, 1025 square feet cooling surface, with 7½″ by 12″ by 12″ horizontal air pump, direct acting, and 6″ centrifugal pump driven by 5″ by 5″ engine	2300
1 surface condenser, 1025 square feet cooling surface; 5″ by 12″ by 10″ Edwards single-cylinder air pump and 6″ centrifugal pump driven by a 5″ by 5″ engine; maximum 28″, referred to 30″ barometer	2850
1 surface condenser, 1025 square feet cooling surface; 6″ by 8″ rotative dry-air pump; 6″ by 6″ Edwards wet-air pump and 6″ centrifugal pump driven by 5″ by 5″ engine; maximum vacuum 29″, referred to 30″ barometer (temp. cooling water 50 degrees F.)	3500

Westinghouse-Leblanc Jet Condenser.

1 jet condenser with turbine-driven pumps, 20,000 pounds steam per hour, 26″ vacuum, 70 degrees F. inlet water	2150
1 jet condenser with turbine-driven pumps, 20,000 pounds steam per hour, 29″ vacuum, 50 degrees F. inlet water	3275

In general the cost of complete condensing equipments installed and ready for operation will approximate as follows:

	Cost per Kilowatt of Main Generating Unit.
Siphon condensers without air pump	$2.00 to $3.00
Jet condensers	3.00 to 4.50
Barometric condensers with dry-air pump	4.00 to 6.00
Surface condensers for 26-inch vacuum	3.50 to 5.00
High-vacuum surface condensers	3.50 to 10.00
Leblanc jet condensers and pumps	2.00 to 6.00

The curve in Fig. 336 shows the relative costs of complete surface condensing plants for steam turbines to maintain the vacua indicated. It will be noted how much more expensive a high-vacuum plant is than one designed for moderate vacua. Thus a 27-inch plant costs 25 per cent more than a 26-inch plant, and a 28.5-inch plant costs twice as much. (J. R. Bibbins, Power, January, 1905.)

The real cost of a condensing plant, however, is not limited to the cost of condensing auxiliaries and piping, but should include all other costs

necessitated by the use of the condensing plant, including cost of extra building space, foundations and the like, and the attending fixed charges.

263. Most Economical Vacuum.* — The load factor, or the ratio of the actual yearly load to the rated yearly capacity, has a marked influence on the degree of vacuum best suited for a given installation, since the fixed charges go on whether the plant is running or not, while the gain due to the higher vacuum is realized only when the engines are operating. The higher the load factor the greater is the amount of power produced, the longer does the apparatus operate at best efficiency, the lower the ratio of fixed charges to total operating expenses, and consequently the lower the cost of power per unit.

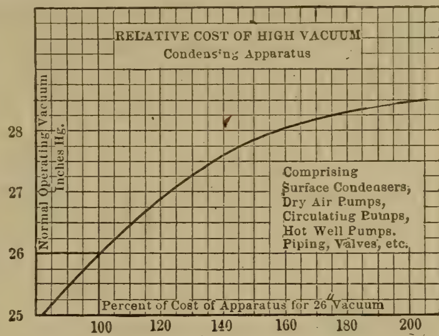


Fig. 336.

The load factor for electric-lighting stations is invariably low and seldom exceeds 35 per cent, with an average not far from 18 per cent. In street-railway work it is higher and averages about 40 per cent. In manufacturing plants the load factor varies considerably, but as a rule is somewhat higher than in either of the above cases. Tables 85 and 86 (Power, December, 1906, p. 769) show the most economical vacua for different load factors for plants of 1000 kilowatts capacity with conditions as stated. From the tables it would seem at first glance that, except where coal is expensive, all the plants with low factors, 10 per cent and under, ought to be run non-condensing. This is true for "one-engine" installations, but not necessarily so where there are a number of engines or turbines. In the latter case higher economy may be effected by providing only a portion of the engines with condensing equipment. The engine carrying the continuous or day load should operate condensing, and the non-condensing engine should carry the peak load. In order that any of the units may be used for the day work, all engines could be connected to the condenser, but only those carrying the day load should be operated condensing. Each installation, of course, must be considered separately and due weight given to the various factors entering into the problem. For an excellent article on the subject see "Condensers for Steam Engines and Turbines," Power, December, 1906, p. 769, and the Engineer, London, April 13, 1906, p. 381; also Proc. A.S.M.E., Oct., 1910, p. 1579.

* See also Elec'n, Lond., Jan. 14, 1910.

264. Choice of Condensers. — The proper selection of a condenser for a proposed installation depends upon the conditions under which the plant is to be operated. When there is a plentiful and cheap supply of good condensing water suitable for boiler feed, and extremely high vacua are not essential, some type of jet condenser will generally be found most desirable. If overhead room permits, a siphon or barometric condenser will probably be most suitable and least expensive.

TABLE 85.

MOST ECONOMICAL VACUUM FOR STEAM TURBINES.

Vacuum referred to 30-Inch Barometer.

Load Factor, per Cent.	Cost of Coal, Dollars per Ton.									
	$1.50		$2.00		$2.50		$3.00		$3.50	
	A	B	A	B	A	B	A	B	A	B
5	N.C.	N.C.	N.C.	N.C.	18	N.C.	20	N.C.	22	N.C.
10	20	N.C.	23	N.C.	25	N.C.	26.5	20	27	22
15	24	17	26.5	20	27	22	27.5	24	27.7	25.8
20	26.5	20	27.3	23	27.6	25.5	27.8	27	27.9	27.5
30	27.5	24	27.8	27	28	27.6	28.1	27.8	28.2	28
50	28	27.6	28.2	27.9	28.3	28	28.4	28	28.5	28

A. Surface-condensing plant; cost $6 per kilowatt of main generator. Fixed charges 12 per cent. Cost of water not included. Rated capacity of generator, 1000 kilowatts.

B. Surface-condensing plant, including cooling towers and extra cost of land, etc.; cost $10 per kilowatt for 26-inch plant, increasing to $14 per kilowatt for 28.5-inch plant. Fixed charges 12 per cent. No charge for water. Rated capacity of generator, 1000 kilowatts.

TABLE 86.

MOST ECONOMICAL VACUUM FOR RECIPROCATING ENGINES.

Vacuum referred to 30-Inch Barometer.

Load Factor, per Cent.	Cost of Coal, Dollars per Ton.									
	$1.50		$2.00		$2.50		$3.00		$3.50	
	A	B	A	B	A	B	A	B	A	B
10	N.C.	N.C.	15	N.C.	18	N.C.	20	N.C.	22	N.C.
15	16	N.C.	20	N.C.	22	N.C.	22.5	16	24	20
20	22.5	N.C.	23	N.C.	23.5	20	24.5	21	25	22
30	24	16	24.5	21	25.5	22	26.4	23	26.8	24
50	25.5	22	26.7	23.5	27.2	23.5	27.5	26.3	27.7	27

A. Surface-condensing plant; cost $7 per kilowatt of main generator. Fixed charge 12 per cent. Cost of water not included. Rated capacity of generator, 1000 kilowatts.

B. Surface-condensing plant, including cooling towers and extra cost of land, etc.; cost $11 per kilowatt for 26-inch plant, increasing to $13 per kilowatt for 27.5-inch plant. Other conditions as in A.

Where there is a plentiful supply of good water for boiler feed but the water which must be used for cooling purposes is very dirty the siphon condenser is preferable to the barometric form. A surface condenser may be used in the latter case if the condensing water is not so dirty as to seriously impair the efficiency by coating the tubes with sediment, and boiler feed water is scarce.

The air-cooled surface condenser is employed only where water of any kind is scarce.

For very high vacua in connection with steam-turbine work the surface condenser is generally adopted, although the barometric condenser in connection with dry-air pumps and the Leblanc type of condenser and pumps are finding favor with many engineers.

In selecting the type of condenser and auxiliaries due weight must be given to the load factor, cost of coal, water, land, building, interest, depreciation and the like, as outlined in the preceding paragraph.

265. Water-Cooling Systems. — When an ample supply of cooling water is unobtainable, for natural or economic reasons, the circulating water may be used over and over again by employing suitable cooling devices. The three most common in practice are

1. The simple cooling pond or tank.
2. The spray fountain.
3. The cooling tower.

266. Cooling Pond. — The water is cooled partly by radiation and conduction but principally by evaporation. The air is seldom saturated normally, and its capacity for absorbing moisture is increased on account of its temperature being raised by contact with the warm water and by radiation. The cooling action is independent of the depth of water and varies directly as the surface, the amount of heat dissipated for each square foot depending upon the temperature of the water, the relative humidity, and the velocity of the air currents. Results of tests are very discordant.

Box in his Treatise on Heat states that the pond surface should approximate 210 square feet per nominal horse power for an engine working twenty-four hours a day. (Treatise on Heat, Box, p. 152.)

If the engine works only twelve hours per day, the area may be reduced to 105 square feet per horse power, because the water will cool during the night, but in that case the depth should be such as to give a capacity of 300 cubic feet per horse power. These figures are based on a reduction in temperature of 122 to 82 degrees F., with air at 52 degrees F. and humidity 85 per cent, the steam consumption per nominal horse power being taken at 62.5 pounds. It appears from tests that

under ordinary conditions, in the northern part of the United States, with engines using 15 pounds of water per horse-power hour and a vacuum of 26 inches, a reservoir having a surface of 120 square feet per horse power would be ample for cooling and condensing water. (W. R. Ruggles, Proc. A.S.M.E., April, 1912, p. 607.)

Box gives the following formula for the rate of evaporation in perfectly calm air:

$$E = (243 + 3.7\,t)\,(V - v), \tag{178}$$

in which

E = evaporation in grains per square foot per hour;
t = temperature of the water, degrees F.;
V = maximum vapor tension in inches of mercury at temperature t;
v = actual vapor tension.

Evaporation is greatly affected by the force of the wind and varies from 2 to 12 times the amount determined from equation (178).

Example: How many pounds of water will be evaporated per square foot per hour from a pond with the temperature of the water and air 80 degrees F.; air perfectly calm; barometric pressure 29.5 inches and relative humidity 70 per cent?

The maximum vapor tension at temperature of 80 degrees is 1.02 inches of mercury. The actual vapor tension will be

$$1.02 \times 0.70\ (= \text{relative humidity}) = 0.714.$$

Substitute these values in equation (178).

$$\begin{aligned} E &= (243 + 3.7 \times 80)\ (1.02 - 0.714) \\ &= 165 \text{ grains per square foot per hour} \\ &= 0.023 \text{ pound per square foot per hour.} \end{aligned}$$

If the temperature of the water were 130 degrees F. and that of the surrounding air 80 degrees F., humidity 70 per cent, the evaporation would be

$$\begin{aligned} E &= (243 + 3.7 \times 80)\ (4.5 - 0.714) \\ &= 2040 \text{ grains per square foot per hour} \\ &= 0.291 \text{ pound per square foot per hour.} \end{aligned}$$

Here 4.5 = maximum vapor tension, corresponding to a temperature of 130 degrees.

267. Spray Fountain. — From equation (178) we see that even under the most favorable circumstances an enormous pond surface is necessary. To facilitate evaporation with a view toward reducing the size of the pond, the hot circulating water is sometimes distributed through pipes and discharged through nozzles, falling to the surface of

the pond in a spray. The following data pertains to the spray-fountain installation at the power plant of the Chattanooga Electric Company, Chattanooga, Tenn. (Street Railway Review, March 15, 1905.)

Adjoining the power house a pond 150 × 300 feet was excavated to a depth of 4 feet, the level of the water being 8 feet below the condensers. Circulating water returned from the condensers is distributed through a set of pipes provided with 42 nozzles through which the water is discharged upwards. The rectangle defined by the center lines of the outermost pipes is 98 feet by 125 feet. The pipes are supported on brick piers spaced at intervals of about 20 feet in each direction. The

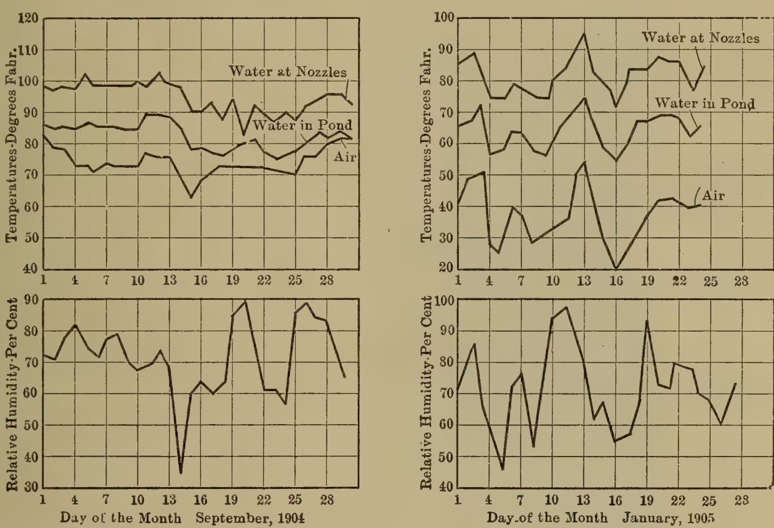


FIG. 337. Curves Showing Performance of Spray Fountain; Chattanooga Electric Company's Power Plant.

discharge opening of the nozzles is $1\frac{1}{4}$ inches in diameter, and the interior is provided with a spiral core so that in its passage the water is given a rotary motion, the effect of which is to greatly increase the spraying action. The nozzles, except on the extreme outer lines of piping, are placed in pairs with the axes in a vertical plane at right angles to the center line of the supply pipe, the axis of each nozzle making an angle of 30 degrees with a vertical plane through the center of the supply pipe. The effect of each pair of nozzles is to throw a mass of spray to the height of about 15 feet, which in falling covers an area of 15 × 30 feet.

A dike extending nearly across the pond near one end provides a canal through which the water is conducted to the suction chamber,

the object being to draw the supply from distant parts of the pond to give greater time for cooling. The "make-up" water is supplied by wells. The operation of the cooling pond for a warm month and for a cold month is shown in Fig. 337. Readings were taken at three-hour intervals. The pond supplies the circulating water for three 2000-square-feet Worthington surface condensers.

Cooling Condensing Water by Means of Spray Nozzles: Power, July 1, 1908, p. 84.

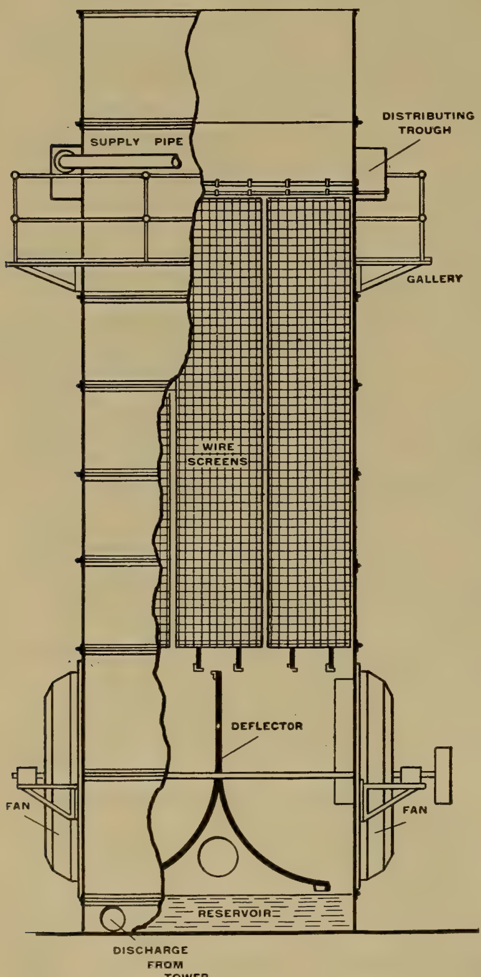


FIG. 338. Barnard-Wheeler Cooling Tower.

268. Cooling Towers. — A cooling tower consists of a wooden or sheet-iron housing open at the top and bottom and so arranged that the heated cooling water may be elevated to the top and distributed in such a manner that it falls in thin sheets or sprays into a reservoir at the bottom, air at the same time being drawn in at the bottom by natural draft or forced in by a fan. The water gives up its heat to the ascending current of air by evaporation and conduction, the latter, however, being a relatively small factor. If the air supply is dependent entirely upon convection, the system is known as the natural-draft or flue cooling tower; if the air is forced into the tower by fans, it is called a fan cooling tower. The different types vary principally in the method of water distribution. Fig. 338 illustrates the Barnard cooling tower, in which the falling water is broken up by vertically suspended galvanized iron wire cloth mats, causing it to trickle in thin sheets to the bottom. A similar result is brought about in the Worthington tower, Fig. 339, by pieces of terra-cotta pipe 6 inches in

diameter and two feet long placed on ends in rows. In the *standard type* of Alberger cooling tower the water trickles down the sides of swamp-cypress boards arranged in honeycomb fashion. In the Alberger *improved type* the fan is placed at the top of the tower with its shaft in a vertical position. The fan is operated by a Pelton water wheel which receives its power from a turbine pump. No oil lubrication is employed, and the operating mechanism is controlled entirely from the engine room. In the Jennison cooling tower the water is divided into a rain of drops, constantly retarded in their fall by a series of perforated 4 × 4-inch galvanized-iron trays arranged in horizontal rows and staggered vertically.

With the best forms of cooling towers, under average conditions, the temperature of the circulating water may readily be reduced from 40 to 50 degrees with a loss not exceeding 3 or 4 per cent of the total quantity of water passing through the tower. The power consumed by the fan in a forced-draft apparatus averages 2 per cent of that developed by the main engines, for the maximum requirements during summer months, and $1\frac{1}{4}$ per cent during the winter.

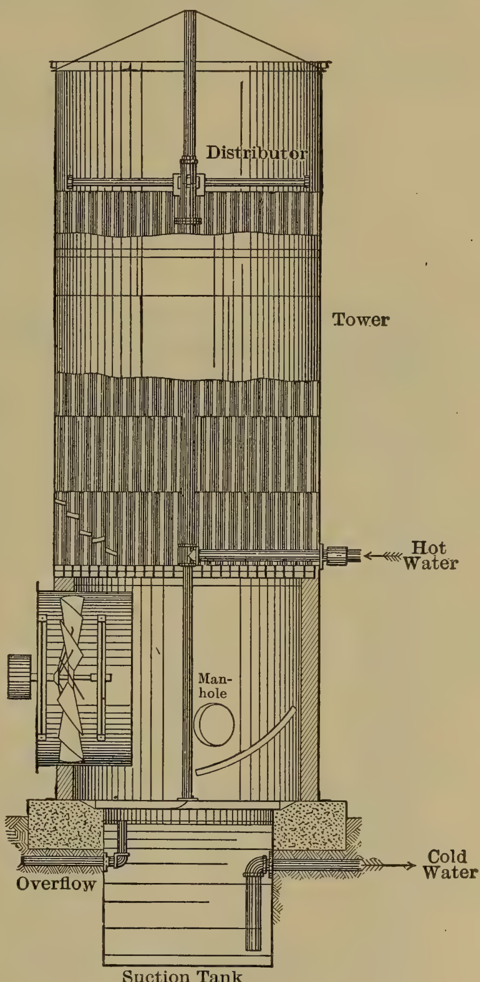


Fig. 339. Worthington Cooling Tower.

The location of the tower may be on the engine-room floor, on top of the building, or in the yard, the latter being the most adaptable. It may be any reasonable distance from the engine and condenser. Fig. 340 shows a typical installation of Worthington condenser and cooling towers.

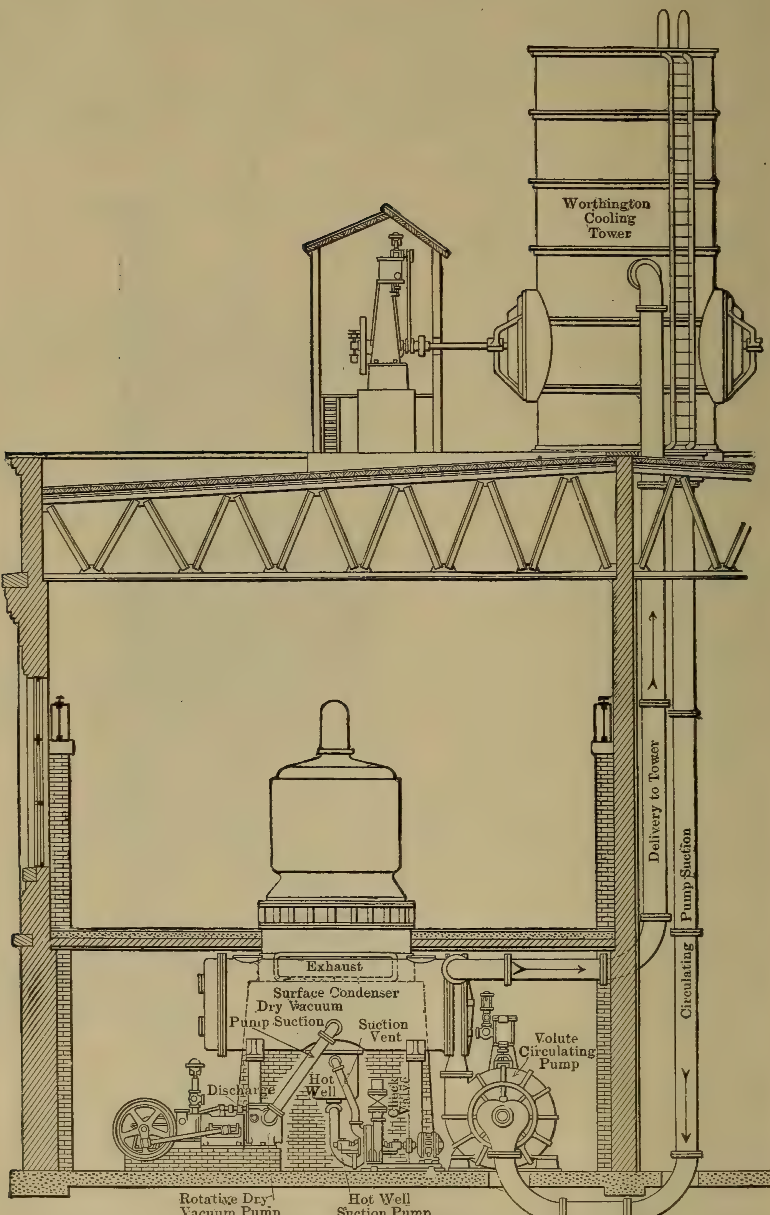


FIG. 340. Typical Cooling Tower Installation.

269. Parallel Comparison of Fan and Natural-draft Cooling Towers.

Fan.	Natural Draft.
Size.	
Small, the forced draft providing sufficient air velocity to effect evaporation.	Large draft being necessarily small, a larger area must be provided to perform same work.
Height limited, because loss from back pressure increases with the height.	Height is an advantage because the tower operates on the principles of a chimney.
Tower usually short and of large area.	
Power Consumption.	
One per cent of station output and upwards, depending upon the type of auxiliaries and the conditions of operation.	None.
Location.	
Inside or outside. Can operate in any location where sufficient head room and air supply are available.	Outside only, unless exceptionally good draft is obtainable.
Especially adapted to inconvenient locations, as roofs, upper decks, boiler floors, etc.	Preferably in the open where advantage may be taken of prevailing winds.
Conditions of Atmosphere.	
Comparatively little affected by temperature, considerably by humidity, and none by winds.	Largely affected by temperature and humidity and wind. Draft increased by steady winds.
Conditions of Operation.	
More especially adapted for heavy continuous duty the year round, as in rail-plants or mills.	Especially adapted for light summer and heavy winter duty, as in electric-lighting plants.
First Cost and Cost of Operation.	
First cost greater on account of mechanical construction and necessary auxiliaries.	First cost small by reason of simplicity and construction.
Cost of operation dependent upon type of auxiliary and conditions of operation.	First cost largely dependent upon materials used in interior construction.
	Cost of operation limited to fixed charges.

Cooling Towers for Steam and Gas Power Plants: Trans. A.S.M.E., Vol. 31, p. 725, 1909.

270. Hygrometry. — The degree of saturation, or relative humidity, is ordinarily determined from the difference in reading of a wet- and a dry-bulb thermometer, thus: If the air is saturated with aqueous vapor no evaporation takes place from the dry bulb and the two thermometers give identical readings; but if it is unsaturated, evaporation occurs.

TABLE 87.

PROPERTIES OF SATURATED AIR. (BAROMETER 29.921.)

Mixture of Air Saturated with Water Vapor.

Temperature, degrees F.	Weight of 1000 Cu. Ft. of Dry Air, Pounds.	Volume of One Lb. of Dry Air, Cu. Ft.	Elastic Force of Vapor, Ins. of Mercury (Marks & Davis).	Elastic Force of the Dry Air in the Mixture, Ins. of Mercury.	Weight of 1000 Cu. Ft., Lbs. Weight of the Dry Air, Content.	Weight of the Vapor, Content.	Total Weight of the Mixture.
1	2	3	4	5	6	7	8
0	86.35	11.58	*0.044	29.88	86.23	0.081	86.31
10	84.51	11.83	*0.069	29.85	84.31	0.125	84.43
20	82.75	12.08	*0.107	29.81	82.44	0.189	82.63
30	81.06	12.33	*0.156	29.76	80.62	0.273	80.89
32	80.73	12.39	0.180	29.74	80.24	0.304	80.54
35	80.24	12.46	0.203	29.72	79.70	0.340	80.04
40	79.43	12.59	0.248	29.67	78.77	0.410	79.18
45	78.64	12.72	0.300	29.62	77.86	0.492	78.35
50	77.88	12.84	0.362	29.56	76.94	0.587	77.53
55	77.12	12.97	0.436	29.48	75.98	0.700	76.68
60	76.38	13.09	0.522	29.40	75.05	0.828	75.88
62	76.08	13.14	0.560	29.36	74.66	0.885	75.54
65	75.65	13.22	0.622	29.30	74.08	0.977	75.06
70	74.94	13.34	0.739	29.18	73.08	1.15	74.23
72	74.65	13.40	0.790	29.13	72.68	1.22	73.90
75	74.24	13.47	0.873	29.05	72.08	1.35	73.42
80	73.55	13.60	1.03	28.89	71.01	1.57	72.58
85	72.87	13.72	1.21	28.71	69.92	1.83	71.75
90	72.21	13.85	1.42	28.50	68.78	2.13	70.91
95	71.56	13.97	1.66	28.26	67.59	2.47	70.06
100	70.92	14.10	1.93	27.99	66.34	2.85	69.19
105	70.29	14.23	2.24	27.69	65.05	3.28	68.33
110	69.67	14.35	2.59	27.33	63.64	3.77	67.41
115	69.07	14.48	2.99	26.93	62.16	4.31	66.47
120	68.47	14.61	3.44	26.48	60.60	4.92	65.52
125	67.88	14.73	3.95	25.97	58.92	5.61	64.53
130	67.31	14.86	4.52	25.40	57.14	6.37	63.51
135	66.74	14.98	5.16	24.76	55.23	7.21	62.44
140	66.19	15.11	5.88	24.04	53.18	8.14	61.32
145	65.64	15.23	6.67	23.25	51.01	9.18	60.19
150	65.10	15.36	7.57	22.35	48.63	10.32	58.95
155	64.57	15.49	8.55	21.37	46.12	11.57	57.69
160	64.05	15.61	9.65	20.27	43.39	12.96	56.35
165	63.54	15.74	10.86	19.06	40.47	14.48	54.95
170	63.04	15.86	12.20	17.72	37.33	16.14	53.47
175	62.54	15.99	13.67	16.25	33.96	17.96	51.92
180	62.05	16.12	15.29	14.63	30.34	19.94	50.28
185	61.57	16.24	17.07	12.85	26.44	22.10	48.54
190	61.09	16.38	19.02	10.90	22.26	24.44	46.70
195	60.63	16.50	21.15	8.77	17.17	27.00	44.77
200	60.17	16.62	23.47	6.45	12.97	29.76	42.73
205	59.71	16.74	26.00	3.92	7.82	32.76	40.58
210	59.27	16.86	28.76	1.16	2.30	35.97	38.27
212	59.09	16.92	29.92	0	0	37.32	37.32

* Regnault.

TABLE 87 (*Continued*).

Ratio of Dry Air to Water Vapor.	Weight of Water Necessary to Saturate 100 Lbs. of Dry Air.	Cu. Ft. of Vapor from One Lb. of Water at Pressure, as in Column 4.	B.t.u. Absorbed by 1000 Cu. Ft. of Dry Air per degree F.†	B.t.u. Absorbed by 1000 Cu. Ft. of Saturated Air per Degree F.‡	Cu. Ft. of Dry Air Warmed One Degree F., per B.t.u.†	Cu. Ft. of Saturated Air Warmed One Degree F., per B.t.u.‡	Temperature Degrees F.
9	10	11	12	13	14	15	16
1064.0	0.094	......	20.51	20.52	48.75	48.74	0
674.0	0.148	......	20.07	20.08	49.80	49.79	10
436.0	0.229	......	19.65	19.67	50.89	50.84	20
295.0	0.338	......	19.25	19.27	51.94	51.89	30
264.0	0.379	3294.0	19.17	19.19	52.16	52.11	32
234.0	0.468	2938.0	19.06	19.08	52.47	52.41	35
192.0	0.521	2438.0	18.86	18.89	53.02	52.94	40
159.0	0.632	2038.0	18.68	18.72	53.53	53.42	45
131.0	0.763	1702.0	18.49	18.54	54.04	53.94	50
108.0	0.921	1430.0	18.31	18.37	54.61	54.43	55
91.0	1.10	1208.0	18.14	18.20	55.12	54.94	60
85.0	1.18	1130.0	18.07	18.13	55.33	55.16	62
76.0	1.32	1024.0	17.96	18.03	55.68	55.53	65
64.0	1.57	871.0	17.80	17.88	56.18	55.93	70
59.0	1.68	817.0	17.74	17.82	56.36	56.12	72
54.0	1.87	743.0	17.63	17.74	56.71	56.37	75
45.0	2.21	637.0	17.47	17.59	57.23	56.85	80
38.0	2.62	546.0	17.31	17.45	57.77	57.30	85
32.0	3.08	469.0	17.14	17.31	58.34	57.77	90
27.0	3.65	405.0	17.00	17.20	58.82	58.14	95
23.0	4.29	351.0	16.84	17.06	59.38	58.62	100
20.0	5.04	305.0	16.69	16.96	59.92	58.96	105
17.0	5.92	266.0	16.54	16.85	60.46	59.35	110
14.0	6.93	232.0	16.40	16.74	60.97	59.74	115
12.0	8.12	203.0	16.26	16.65	61.50	60.06	120
11.0	9.46	178.0	16.12	16.57	62.03	60.35	125
9.0	11.20	157.0	15.98	16.50	62.58	60.61	130
7.7	12.90	139.0	15.85	16.41	63.09	60.93	135
6.5	15.30	123.0	15.72	16.37	63.61	61.08	140
5.5	17.90	109.0	15.59	16.33	64.14	61.23	145
4.7	21.70	96.9	15.46	16.29	64.68	61.38	150
4.0	25.10	86.4	15.33	16.27	65.23	61.46	155
3.3	29.90	77.2	15.21	16.26	65.75	61.50	160
2.8	35.80	69.1	15.09	16.27	66.27	61.46	165
2.3	43.20	62.0	14.97	16.29	66.80	61.38	170
1.9	54.80	55.7	14.85	16.33	67.34	61.23	175
1.5	65.70	50.1	14.74	16.38	67.84	61.05	180
1.2	83.80	45.2	14.62	16.44	68.40	60.83	185
0.91	111.0	40.9	14.51	16.53	68.92	60.49	190
0.66	191.0	37.0	14.40	16.64	69.44	60.96	195
0.44	229.0	33.6	14.29	16.77	69.98	59.63	200
0.24	419.0	30.5	14.18	16.92	70.52	59.10	205
0.06	......	27.8	14.07	17.09	71.07	58.58	210
0	......	26.8	14.03	17.16	71.33	58.27	212

† Mean specific heat of air taken as 0.2375. ‡ Mean specific heat of water vapor taken as 0.46.

The wet-bulb thermometer is thus cooled and its readings are lower than those of the dry-bulb. The difference in reading is a function of the relative humidity, and the latter may be calculated from the following rational psychrometric formula deduced by Willis H. Carrier (Proc. A.S.M.E., Nov., 1911).

$$h = \left[P_w - \frac{(P - P_w)d}{2800 - 1.3\,t_w}\right]\frac{100}{P_d}, \tag{179}$$

in which

h = relative humidity, per cent;

P_w = maximum tension of aqueous vapor corresponding to the temperature of the wet-bulb thermometer, inches of mercury; (This may be taken directly from Steam Tables.)

P = observed barometric pressure, inches of mercury;

d = difference in reading of the wet and dry-bulb thermometers, degrees F.;

t_w = temperature of the wet-bulb thermometer;

P_d = maximum tension of aqueous vapor corresponding to the temperature of the dry-bulb thermometer, inches of mercury.

Example: Determine the relative humidity when the dry bulb reads 70 degrees F., wet bulb 60 degrees F., barometer 28.0.

From the Steam Tables we find

$$P_w = 0.522 \qquad P_d = 0.739$$

whence

$$h = \left[0.522 - \frac{(28 - 0.522)10}{2800 - 1.3 \times 60}\right]\frac{100}{0.739}$$

$$= 57.0 \text{ per cent.}$$

If the relative humidity h, at barometric pressure P, is to be referred to barometric pressure P_1, the relative humidity under the new pressure will be

$$h_1 = \frac{P_1 h}{P}. \tag{180}$$

Example: Required the relative humidity in the preceding problem if the air and vapor content are compressed to 30 inches

$$h_1 = \frac{30 \times 57}{28} = 61 \text{ per cent.}$$

The weight of moisture per cubic foot of mixture is

$$w = hy, \tag{181}$$

in which y = weight of one cubic foot of water vapor at temperature corresponding to the dry-bulb thermometer. (This is the density of steam at the specified temperature and may be taken directly from Steam Tables.)

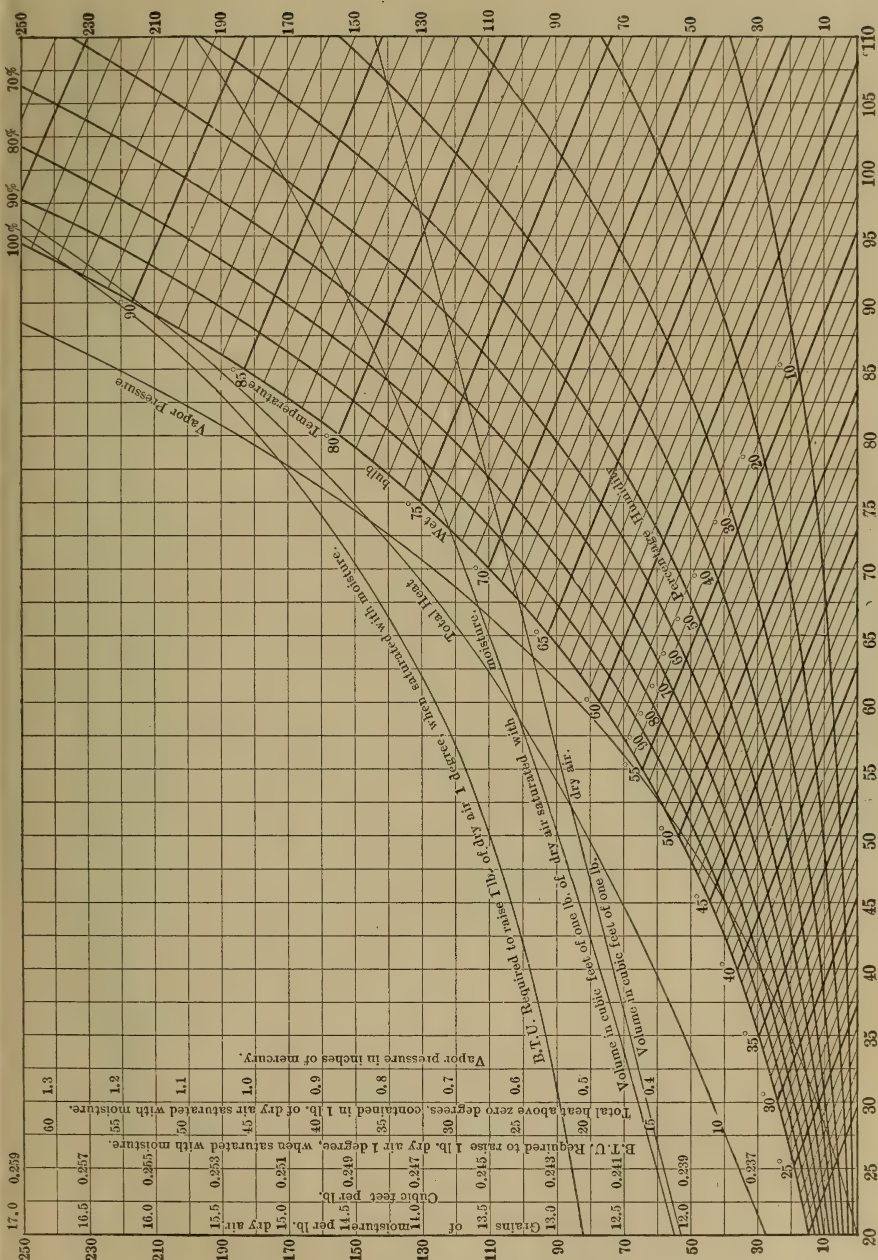


FIG. 341. Psychrometric Chart (W. H. Carrier).

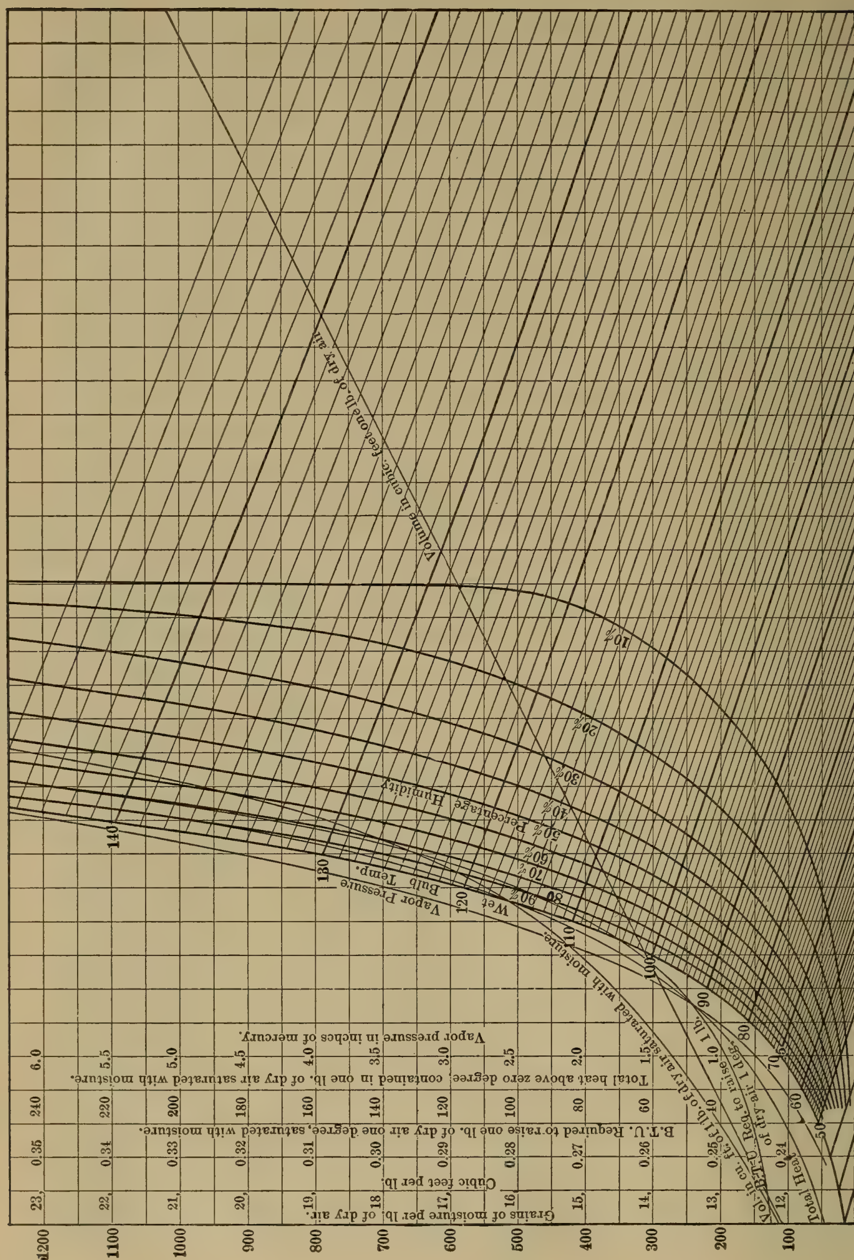


Fig. 342. Psychrometric Chart (W. H. Carrier).

Table 87 gives the properties of saturated air for various temperatures in terms of volumes of the mixture and is taken from the author's paper "Properties of Saturated and Unsaturated Air," Jour. Franklin Inst., Feb., 1911.

Figs. 341 and 342 are taken from Willis H. Carrier's paper, "Rational Psychrometric Formulæ," Proc. A.S.M.E., Nov., 1911, p. 1309, and give the various properties of saturated and unsaturated air for various temperatures and relative humidities in terms of one pound of dry air.

271. Water-cooling Calculations. — Air is said to be completely saturated when it contains all the water vapor it can hold without causing precipitation. If the vapor content is less than that corresponding to complete saturation the air will tend to become saturated by absorbing moisture from surrounding objects. The drier the air the greater will be its affinity for moisture. The necessary latent heat for vaporization is supplied directly by the water producing the vapor or by the surrounding objects in contact with the water. Thus, in the open cooling tower the water vapor is absorbed from the circulating water, and the heat necessary to effect this vaporization is given up by the water, with a resultant reduction in temperature of the water itself; and in the evaporative surface condenser the vapor is absorbed from the water spray in contact with the tubes, the heat required to effect this vaporization being given up by the steam within the condenser chambers, resulting in condensation of the steam. If the air coming in contact with the water is very dry and at a high temperature the vaporization of the water may be rapid enough to cool the remaining water to a temperature much lower than that of the air. In this case practically all of the cooling is effected by *evaporation*. But when the air is at a low temperature and high relative humidity a considerable amount of heat may be carried away by the air by *conduction*. The *quantity* of air and water necessary to produce a given cooling effect may be determined as follows:

Let H = total amount of heat to be abstracted, B.t.u. per hour;
W = weight of water to be cooled, pounds per hour;
t_e = temperature of water entering cooling device;
t_l = temperature of water leaving cooling device;
t_0 = temperature of air entering cooling device, degrees F.; $T_0 = t_0 + 460$;
t_2 = temperature of air leaving cooling device, degrees F.; $T_2 = t_2 + 460$;
p = ordinary atmospheric pressure = 29.92 inches of mercury;
p_a = observed atmospheric pressure, inches of mercury;

p_0 = elastic force of vapor at temperature t_0, inches of mercury;
p_2 = elastic force of vapor at temperature t_2, inches of mercury;
V_0 = volume of air entering the cooling device, cubic feet per hour, atmospheric conditions;
V_2 = volume of air discharged from the cooling device at temperature t_2;
d = density of *dry* air, at pressure p and temperature t_0;
h_0 = weight of moisture in 1 cubic foot of saturated air at temperature t_0, pounds;
h_2 = weight of moisture in 1 cubic foot of saturated air at temperature t_2 pounds;
z_0 = relative humidity of the air entering the cooling device;
z_2 = relative humidity of the air leaving the cooling device;
C = mean specific heat of dry air at constant pressure = 0.2375;*
S = mean specific heat of water vapor at temperature t_2;
r_2 = heat of vaporization at temperature t_2.

The *pressure* p_1 of the *dry* air in atmospheric air entering the cooling device is

$$p_1 = p_a - p_0 z_0. \tag{182}$$

The *pressure* p_3 of the *dry* air leaving the cooling device is

$$p_3 = p_a - p_2 z_2. \tag{183}$$

The *weight* w of *dry* air entering the cooling device under atmospheric temperature and pressure, pounds per hour, is

$$w = \frac{p_1}{p} d V_0. \tag{184}$$

The *weight* w_0 of *moisture* carried into the cooling device by the air, pounds per hour, is

$$w_0 = h_0 z_0 V_0. \tag{185}$$

The *volume* of *air leaving* the cooling device is

$$V_2 = V_0 \frac{p_1}{p_3} \cdot \frac{T_2}{T_0}. \tag{186}$$

The *weight* w_2 of *moisture* carried away by the air *discharged* from the cooling device, pounds per hour, is

$$w_2 = h_2 z_2 V_2. \tag{187}$$

The *weight* of *circulating water* w_3 *absorbed by the air in passing through the cooling device,* pounds per hour, is

$$w_3 = w_2 - w_0. \tag{188}$$

* The mean specific heat of air varies with the temperature, see Fig. 7.

The *heat* H to be *abstracted* from the *circulating water*, B.t.u. per hour, is

$$H = W(t_l - t_e). \tag{189}$$

The heat is dissipated, by the cooling process, in raising the temperature of the air and vapor entering the cooling device from t_0 to t_2 (by *conduction*) and in evaporating the moisture absorbed by the air in passing through the apparatus (by *evaporation*).

The *heat* H_a required to raise the temperature of dry air from t_0 to t_2, B.t.u. per hour, is

$$H_a = Cw(t_2 - t_0). \tag{190}$$

The *heat* H_s required to superheat the water vapor entering the cooling device from temperature t_0 to t_2, B.t.u. per hour, is

$$H_s = w_0S(t_2 - t_0). \tag{191}$$

The *heat* H_c abstracted by *conduction* from the circulating water, B.t.u. per hour, is

$$H_c = H_a + H_s. \tag{192}$$

The *heat* H_e abstracted from the circulating water by *evaporation*, B.t.u. per hour, is

$$H_e = w_3r_2. \tag{193}$$

Though the process of evaporation is practically continued through the whole range in the cooling device, we are justified in using the heat of vaporization at the highest temperature, because the liquid was at this temperature entering the cooling device and the vapor is brought back to the temperature when leaving it.

The *total heat* H_t absorbed by the air in passing through the cooling device, B.t.u. per hour, is $H_t = H_c + H_e$. (194)

Neglecting radiation and other minor losses, the heat H_t absorbed by the air must be equal to the heat given up by the circulating water, or

$$H_t = H. \tag{195}$$

Example: Determine the quantity of air passing through the cooling tower per hour and the circulating water lost by evaporation in a power plant operating under the following conditions: Engines indicate 500 horse power and consume 20 pounds steam per i.h.p. hour; temperature of the injection water, discharge water and outside air, 90, 122, and 72 degrees F., respectively; barometer 29.5; relative humidity of air entering and leaving tower 70 and 90 per cent respectively; vacuum at condenser 25 inches. Determine also the weight of water evaporated in per cent of that circulated and of the condensed steam.

In the problem,

$p_a = 29.5$,	$t_0 = 72$,	† $h_0 = 0.001224$,
† $p_0 = 0.79$,	* $t_2 = 112$,	† $h_2 = 0.003978$,
† $p_2 = 2.79$,	$t_e = 122$,	$z_0 = 0.70$,
$d = 0.0747$,	$t_l = 90$,	$z_2 = 0.90$,
$C = 0.2375$,	$S = 0.45$,	$r_2 = 1028.9$.

These values are obtained from Steam Tables and from Air Tables (Table 87).

Substitute these values in equations (182) to (195) thus:

$$(182),\quad p_1 = 29.5 - 0.79 \times 0.7 = 28.95.$$

$$(183),\quad p_3 = 29.5 - 2.74 \times 0.9 = 27.03.$$

$$(184),\quad w = \frac{28.95}{29.92} \times 0.0747\, V_0 = 0.0722\, V_0.$$

$$(185),\quad w = 0.001224 \times 0.7\, V_0 = 0.000857\, V_0.$$

$$(186),\quad V_2 = \frac{28.95}{27.03} \cdot \frac{460 + 112}{460 + 72} V_0 = 1.152\, V_0;$$

that is, each cubic foot of dry air entering the cooling-tower is increased in volume to 1.152 cubic feet as it leaves.

$$(187),\quad w_2 = 0.003978 \times 0.9 \times 1.152\, V_0 = 0.004125\, V_0.$$

$$(188),\quad w_3 = 0.004125\, V_0 - 0.000857\, V_0 = 0.003268\, V_0.$$

The total heat to be abstracted from the steam is

$$H = 500 \times 20\, (1120.1 - 122 + 32) = 10{,}300{,}000 \text{ B.t.u. per hour.}$$

(189), But $W\,(122 - 90) = 10{,}300{,}000$,

from which $W = 322{,}000$ pounds per hour.

$$(190),\quad H_a = 0.2375 \times 0.0722\, V_0\, (112 - 72) = 0.6865\, V_0.$$

$$(191),\quad H_s = 0.000857\, V_0 \times 0.45\, (112 - 72) = 0.001543\, V_0.$$

* By assumption, t_2 being 10 to 20 degrees lower than t_e in average practice when the range $t_e - t_o$ is greater than 30 degrees.

† Marks and Davis.

$$\begin{aligned}(192),\quad H_c &= 0.6865\ V_0 + 0.001543\ V_0\\ &= 0.688\ V_0.\\ (193),\quad H_e &= 0.003268\ V_0 \times 1028.9\\ &= 3.365\ V_0.\end{aligned}$$

$$\frac{H_e}{H_c} = \frac{3.365\ V_0}{0.688\ V_0} = 4.89;$$

that is, the *air removes* 4.89 *times more heat by evaporation than by conduction* under the given conditions.

$$\begin{aligned}(194),\quad H_t &= 0.688\ V_0 + 3.365\ V_0\\ &= 4.053\ V_0.\end{aligned}$$

$$\frac{H_s}{H_t} = \frac{0.001543\ V_0}{4.053\ V_0} = .00038;$$

that is, the heat required to superheat the moisture carried into the tower by the air is approximately $\frac{4}{100}$ of 1 per cent of the total; hence an error as great as 20 per cent in the mean specific heat of the vapor is negligible.

$$(195),\quad 4.053\ V_0 = 10{,}300{,}000,$$

from which

V_0 = 2,543,000 cubic feet of air per hour necessary to effect the required cooling.
= 42,300 cubic feet per minute.

From (188)

$$w_3 = 0.003268\ V_0.$$

Substitute

V_0 = 2,543,000 in above equation.
$w_3 = 0.003268 \times 2{,}543{,}000$
= 8320 pounds, or the weight of circulating water carried away per hour.

$$\frac{w_3}{W} = \frac{8320}{322{,}000} = .0258;$$

that is, 2.58 *per cent of the circulating water is carried away by the air in effecting the necessary cooling.*

$$\frac{w_3}{20 \times 500} = \frac{8320}{10{,}000} = .832;$$

that is, the equivalent of 83.2 per cent of the steam used by the engines is evaporated in the cooling tower, or the *make-up water is more than supplied by the condensed steam.*

Example: Evaporation Surface Condenser. — How many cubic feet of air and how many pounds of water spray must be forced through an evaporative surface condenser of the fan type in order to condense 1000 pounds of steam per hour and maintain a vacuum of 25 inches, barometer 29? (Atmospheric air 80 degrees F., relative humidity 70 per cent.) The air and vapor issue from the discharge pipe under pressure of 4 inches of water, temperature 120 degrees F., relative humidity 98 per cent.

The absolute pressure in the condenser is 29.0 − 25.0 = 4 inches of mercury.

The total heat to be withdrawn in order to cool and condense 1000 pounds of steam per hour at absolute pressure of 4 inches to 120 degrees F. is

$$1000\,[1114.8 - (120 - 32)] = 1{,}026{,}000 \text{ B.t.u.}$$

Neglecting radiation and leakage losses, this is the heat to be abstracted per hour by the air and water spray.

The pressure of the dry air in the mixture entering the condenser is, equation (182),

$$\begin{aligned} p_1 &= 29.0 - 0.7 \times 1.029 \\ &= 28.28. \end{aligned}$$

The pressure of dry air in the mixture leaving the condenser is, equation (183),

$$\begin{aligned} p_3 &= (29.0 + 0.294) - 0.98 \times 3.438 \\ &= 25.925 \end{aligned}$$

(0.294 is the value in inches of mercury of 4 inches of water-fan pressure).

Let V_0 = volume of atmospheric air *entering* the condenser. The volume leaving the condenser will be, equation (186),

$$V_2 = \frac{28.280}{25.925} \cdot \frac{460 + 120}{460 + 80} V_0 = 1.172\, V_0.$$

The weight of vapor in the condenser discharge is, equation (187),

$$\begin{aligned} w_2 &= 1.172\, V_0 \times 0.004888 \times 0.98 \\ &= 0.005615\, V_0 \text{ pounds.} \end{aligned}$$

The weight of vapor in the mixture entering the condenser is, equation (185),

$$\begin{aligned} w_0 &= 0.00157 \times 0.7\, V_0 \\ &= 0.001099\, V_0 \text{ pounds.} \end{aligned}$$

The amount evaporated therefore is

$$\begin{aligned} w_3 &= 0.005615\ V_0 - 0.001099\ V_0 \\ &= 0.004516\ V_0 \text{ pounds.} \end{aligned}$$

The *weight* of *dry air* entering the condenser is, equation (184),

$$\begin{aligned} w &= \frac{28.28}{29.921}\ 0.07362\ V_0 \\ &= 0.06958\ V_0 \text{ pounds.} \end{aligned}$$

The *heat* absorbed by the *dry air* in being heated from 80 degrees to 120 degrees F. is, equation (190),

$$\begin{aligned} H &= Cw\ (t_2 - t_0) \\ &= 0.2375 \times 0.06958\ V_0\ (120 - 80) \\ &= 0.658\ V_0 \text{ B.t.u.} \end{aligned}$$

Heat required to superheat w_0 pounds of vapor from 80 degrees to 120 degrees F. is, equation (191),

$$\begin{aligned} H_0 &= 0.001099\ V_0 \times 0.46\ (120 - 80) \\ &= 0.02022\ V_0 \text{ B.t.u.} \end{aligned}$$

Heat absorbed by the evaporation of w_3 pounds of water is, equation (193),

$$\begin{aligned} H_e &= 0.004516\ V_0 \times 1046.7 \\ &= 4.720\ V_0 \text{ B.t.u.} \end{aligned}$$

(Here the latent heat is taken at the lower temperature, it being the original temperature of the liquid.)

Total heat absorbed by the entering air and spray is

$$\begin{aligned} H_t &= 0.658\ V_0 + 4.720\ V_0 + 0.020\ V_0 \\ &= 5.398\ V_0. \end{aligned}$$

But this represents also the heat given up by the steam, or

$$5.398\ V_0 = 1{,}026{,}000,$$

from which $V_0 = 190{,}500$ cubic feet of atmospheric air necessary to condense and cool 1000 pounds of steam under the given conditions.

The water spray to be injected per hour is

$$0.004516\ V_0 = 0.004516 \times 190{,}500 = 860 \text{ pounds.}$$

272. Test of Cooling Towers. — The following gives the results of a test made on the cooling-tower plant of the A. F. Brown Company at Elizabethport, N. J. The tower is working in connection with a Wheeler surface condenser of 280 square feet of cooling surface, mounted over a 10, 12 × 12 combined air and circulating pump.

Observations made on June 24, 1904.

Temperature of air	81 degrees
Hygrometer	69 degrees
Temperature of air at top of tower	89 degrees
Temperature of water in troughs	105 degrees
Temperature of water in tank	83 degrees
Revolutions of fan, 239 r.p.m., air pressure	$\frac{3}{8}$ inch water
Velocity of air out of tower	822 feet per minute
Gallons of water passing over mats	385 per minute
Vacuum	26 inches
Temperature of air-pump discharge	87 degrees

Observations made June 28, 1904, 9 A.M.

Temperature of air	76 degrees
Hygrometer	59 degrees
Temperature of air at top of tower	81 degrees
Temperature of water in troughs	96 degrees
Temperature of water in tank	78 degrees
Revolutions of fan, 232 r.p.m., air pressure	$\frac{3}{8}$ inch water
Velocity of air out of tower	680 feet per minute
Gallons of water passing over mats	406 per minute
Vacuum	25.5 inches
Temperature of air-pump discharge	90 degrees

Observations made June 28, 1904, 3 P.M.

Temperature of air	74 degrees
Hygrometer	57 degrees
Temperature of air at top of tower	83 degrees
Temperature of water in troughs	99 degrees
Temperature of water in tank	80 degrees
Revolutions of fan, 237 r.p.m., air pressure	$\frac{5}{16}$ inch water
Velocity of air out of tower	769 feet per minute
Gallons of water passing over mats	470 per minute
Vacuum	25.5 inches
Temperature of air-pump discharge	92 degrees

Observations made June 29, 1904.

Temperature of air	78 degrees
Hygrometer	71 degrees
Temperature of air at top of tower	86 degrees
Temperature of water in troughs	108 degrees
Temperature of water in tank	82 degrees
Revolutions of fan, 241 r.p.m., air pressure	$\frac{3}{8}$ inch
Velocity of air out of tower	772 feet per minute
Gallons of water passing over mats	430 per minute
Vacuum	25.5 inches
Temperature of air-pump discharge	93 degrees

RESULTS OF TEST OF NATURAL-DRAFT TOWER, DETROIT.

COMPLETE FIVE-FIFTHS SURFACE INSTALLED.

Proc. A.S.M.E., Mid-Nov., 1909, p. 1205.

Engines:	Two 400-i.h.p. 300-kw. MacIntosh & Seymour tandem-compound engines, overhung generators.
Condensers:	Worthington surface (admiralty type) 1600-sq. ft. reciprocating wet-air pump and circulating pump.
Tower:	Wood-mat construction, 24,500 sq. ft. evaporating surface, exclusive of shell.
Test:	March 15 to 16, 1901, 4 p.m. to 4 p.m., 24 hr.

		A.M.		P.M.	AVERAGE.
Weather:	Barometer (abs.), min.......	30.22	30.07;	30.14	30.27
	Temperature air, deg........	18.5	25;	30	25
	Relative humidity, per cent .	76	82;	58	72

Load:	600 kw. max. to 50 kw. min. Average	244.9 kw.
	Engine efficiency = 92.5 = 875 i.h.p. max. Average ..	354.8 i.h.p.
Steam:	Weight of condensed steam per hr., lb.................	5910.6
	Temperature exhaust steam, deg. F....................	134.38
	Temperature condensed steam, deg. F.................	108.78
	Weight of steam per hour, max. load, lb.............	13,500
	Vacuum (abs.) 25 to 19, average about................	22
	Vacuum corresponding to temperature exhaust steam...	25
	Vacuum possible with good condenser (10 deg. difference)	28
Water:	Circulated per hr., lb..............................	293,536
	Temperature hot well, average, deg. F.................	87.50
	Temperature cold well, average, deg. F................	71.27
	Vaporization loss per hr., lb.........................	5970
Results:	Condenser surface per kw., sq. ft.....................	2.66
	Steam per kw. hr., lb.................................	24.3
	Steam per i.h.p. hr., lb..............................	16.66
	Circulating water per lb. of steam, lb.................	49.6
	Steam per sq. ft. condenser surface per hr., lb.........	3.7
	Circulating water per sq. ft. tower surface, lb..........	12
	Difference in temperature between exhaust steam and discharge, deg. F..................................	47
Cooling:	Max. 20 deg., min. 3 deg.-5 deg. Average.............	16.23
	Heat dissipated per hr., B.t.u........................	4,769,000
	Heat per sq. ft. tower surface, B.t.u..................	195
	Heat per sq. ft. per 1000 lb. water, B.t.u..............	0.665
Evaporation:	Circulating water, per cent	2.03
	Engine steam, per cent...............................	101
Tower:	Surface per kw. (average load 245 kw.), sq. ft..........	100
	Surface per kw. (max. load 600 kw.), sq. ft............	40.8
	Surface per 1000 lb. steam max. load, sq. ft.	1820
	Surface per 1000 lb. steam average load, sq. ft.........	4140
	Surface per 1000 lb. circulating water per deg. max. cooling, sq. ft.......................................	4.17

Time.	Temperature, Deg. Fahr.					Quantities.				
	Air.	Hot Well.*	Cold Well.	Water Cooling.	Total Heat Head.†	Tower Water, Lb. per Hr.	Heat Dissipated, B.t.u. Lb. per Hr.	Heat per Sq. Ft. Cooling Surface, B.t.u. per Hr.	Circulating Water per Sq. Ft., Lb. per Hr.	Load, Kw.
1	2	3	4	5	6	7	8	9	10	11
12 noon	34	102	89	13	68	375,000	4,880,000	332	25	270
1.30	35	106.5	90	16.5	71.5	‡ 375,000 370,200	6,108,000	415	24.8	315 290
2.30	35	106.5	87.5	19	71.5	375,000	7,120,000	484	25	315
3.30	35	113	88.5	24.5	78	375,000	9,000,000	613	25	350
4.30	32.5	100	84	16	67.5	399,000	6,384,000	434	26.6	365
5.00	28.5	103.5	88	15.5	75	445,500	6,900,000	470	29.7	485
6.00	26	125	94	31	99	417,000	12,930,000	880	27.8	655
7.00	24	121	94	27	97	427,000	11,532,000	785	27.4	570
8.00	24	123	94.5	28.5	99	427,000	12,174,000	827	27.4	600

* Assuming a more efficient condenser, say 10 deg. difference, the probable vacuum would be 26 deg. to 27.5 deg. This condenser actually operated at 40 deg. to 50 deg. difference.

† Total heat head = air heating + lost head. ‡ Difference due to rapid change in load.

CHAPTER XII.

FEED-WATER PURIFIERS AND HEATERS.

273. General. — All natural waters contain more or less foreign matter either in suspension or solution. Waters containing carbonates and sulphates of magnesia and lime, soluble salts of silica, iron, and alumina, and suspended matter tend to form scale in the boiler and reduce its steam-generating capacity and economy. The loss due to this cause is often overestimated but is of secondary importance to the danger due to retarded heat transmission which overheats and weakens the plates and tubes.

Table 88 gives the results of a number of tests made on locomotive boiler tubes with different thicknesses and characters of scale. The diversity of the results indicates the futility of basing the decrease in conductivity on the thickness of the scale. For example, test No. 1 shows a decrease in conductivity of 9.1 per cent for a scale 0.02 inch thick, while No. 16 shows a decrease of only 6.75 per cent for a scale over 6.5 times as thick. The scale in each case was even, hard, and dense. Again, No. 8 with a very soft scale 0.042 inch thick gives a decrease in conductivity of 9.54 per cent, whereas No. 14, also very soft but twice as thick, gives a decrease of only 4.95 per cent. No doubt the heat transmission is a function of the chemical as well as the physical properties, but further experiments are necessary before any specific conclusion can be drawn.

Waters containing acids, organic matter, and magnesium chloride and sulphate tend to corrode the boiler, and those containing sodium carbonate, organic matter, and alkalies induce priming. Even distilled water, as obtained from a surface condenser, is a solvent of iron to a certain extent and causes corrosion and pitting. Table 89 gives some idea of the character and extent of impurities in water from various localities, with an analysis of the scale produced by the water and the trouble in the boiler arising from its use.

It is impossible to judge the quality of feed water merely by the grains of solids per gallon since a large amount of soluble salt such as sodium chloride will not be as deleterious as a very small amount of calcium sulphate.

TABLE 88.

INFLUENCE OF SCALE ON HEAT TRANSMISSION.

(Locomotive Boiler Tubes.)

No.	Thickness of Scale, Inches.	Character of Scale.	Decrease in Conductivity due to Scale. Per cent.
1	.02	Hard, dense	9.1
2	.02	Hard	2.02
3	.033	Soft	4.3
4	.033	Very hard	3.5
5	.038	Medium	4.03
6	.04	Soft, porous	6.82
7	.04	Hard, dense	3.07
8	.042	Very soft	9.54
9	.047	Hard	2.75
10	.065	Medium	2.39
11	.07	Soft	2.38
12	.07	Hard	4.43
13	.085	Soft, porous	19.0
14	.089	Very soft	4.95
15	.11	Hard, porous	16.73
16	.13	Hard, dense	6.75

From tests conducted at the University of Illinois, *Railroad Gazette*, Jan. 27, 1899, June 14, 1901. See also *Engineering Record*, Jan. 14, 1905, p. 53; *Power*, February, 1903, p. 70; *Street Railway Review*, July 15, 1901, p. 415.

The following is a rough rating according to the number of grains of incrusting solids per United States gallon:

Less than
8 grains ....very good.
12 to 15 grains ....good.
15 to 20 grains ....fair.
20 to 30 grains ....bad.
Over 30 grains ....very bad.

This applies to calcium carbonate, magnesium carbonate, and magnesium chloride. For water containing sulphate of calcium and magnesium, divide the first column by 4 for the same rating.

On account of the great variety of possible impurities the proper treatment to be adopted can be determined only by chemical analysis of the feed water in each case.

Table 90, compiled by the Hartford Steam Boiler Inspection and Insurance Company, shows the number of boilers inspected by that company during the year 1911 and the number found defective from various causes.

TABLE 89.

WATER AND BOILER SCALE ANALYSES.

Water Analysis. Grains Per U.S. Gallon.	Lake Michigan, Chicago.	Well 115 Feet, St. Bernard, Ohio	San Francisco, Cal.	Schuylkill River, Philadelphia.	Camaguey, Cuba.	Park City, Utah.	Toledo, Ohio.	Kewanee, Ill.	Arkansas River, Florence, Colo.	Surface Water, Auburn Park, Ill.
	1	2	3	4	5	6	7	8	9	10
Silica	0.438	0.677	0.759	0.338	2.873	1.354	0.759	0.373	0.630	0.508
Oxide of iron and aluminum	0.099	0.116	0.116	0.093	0.140	0.350	0.163	0.081	0.075	0.175
Carbonate of lime	3.731	2.271	4.207	0.068	10.270	1.476	5.519	1.721	2.158	2.382
Sulphate of lime	0.962	4.083	0.680	2.257	3.220	1.360	3.950	1.360	18.540	3.154
Carbonate of magnesia	2.092	4.424	2.866	0.884	4.900	0.318	2.592	2.212	4.848	2.875
Sodium and potassium sulphates	Trace	Trace	1.681	Trace	Trace	0.867	Trace	12.928	11.319	Trace
Sodium and potassium chlorides	0.670	0.990	2.970	0.990	5.708	1.980	2.740	26.070	2.028	1.650
Organic matter	0.066	0.584	....	0.700	33.000	2.569	1.052	0.584	0.701	0.584
Total mineral matter	8.058	12.614	13.665	4.672	32.288	7.826	15.885	45.318	40.062	11.096
Chloride of magnesia	....	....	....	....	5.6	......	....	....	....	....

SCALE ANALYSIS — PER CENT.

	1	2	3	4	5	6	7	8	9	10
Character of sample	Hard, brittle.	Medium hardness	Hard, brittle.	Hard, impervious	Very hard.	Very hard.	Hard.	Soft, brittle.	Hard, crystalline.	Medium hardness
Silica	20.60	8.44	11.18	12.30	24.42	19.00	4.96	2.52	6.20	5.7
Oxide of iron and aluminum	10.30	1.30	10.44	6.18	1.02	6.26	11.80	4.92	2.36	2.04
Carbonate of lime	33.86	37.22	40.96	21.26	29.10	29.02	3.74	18.18	18.78	29.86
Sulphate of lime	None	33.82	Trace	34.62	0.96	5.48	55.38	54.76	59.84	39.64
Carbonate of magnesia	6.04	Trace	22.60	Trace	Trace	Trace	8.19	Trace	0.84	Trace
Magnesia (MgO)	15.48	12.01	....	8.20	25.94	1.45	6.86	9.08	4.75	13.8
Moisture and organic matter	12.89	6.22	13.58	11.70	16.66	13.69	8.69	7.40	5.73	7.64
Oil	Trace	....	....	0.27	....	......	....	2.92	....	....
Loss and undetermined	0.83	0.99	1.24	0.23	1.90	1.55	0.38	0.22	1.50	1.32
Lime (CaO)	....	....	....	5.24	....	23.55	....	....	....	....

1. This water will cause the deposit of a moderate amount of scale which will be hard and persistent.

2. This water will cause a large amount of scale to deposit.

3. This water will cause a moderate amount of scale with a decided tendency to galvanic action on account of the large proportion of sodium and potassium salts present.

4. This water will cause the formation of a moderate amount of very hard scale.

5. This water will cause the deposition of a moderate amount of hard scale. The sodium and potassium salts together with the chloride of magnesia will induce galvanic action with consequent corrosion, pitting, etc.

6. This water will cause the formation of some scale. There is also a decided tendency to corrosive action.

7. Will cause a hard and impervious scale to form.

8. Will cause formation of some incrustation of medium hardness. It will also cause considerable trouble due to galvanic action, foaming and priming.

9. This is not a desirable feed water. It will cause the formation of considerable scale and will cause corrosion, pitting, and possibly foaming.

10. Will cause the formation of a moderate amount of very hard scale.

TABLE 90.

SUMMARY OF INSPECTOR'S REPORTS FOR THE YEAR 1911.

(Hartford Steam Boiler Inspection and Insurance Company.)

Nature of Defects.	Whole Number.	Dangerous.
Cases of deposit of sediment	19,471	1,376
Cases of incrustation and scale	43,663	1,468
Cases of internal grooving	2,830	229
Cases of internal corrosion	13,781	611
Cases of external corrosion	9,668	801
Defective braces and stays	2,611	524
Settings defective	5,677	687
Furnaces out of shape	7,674	402
Fractured plates	3,654	521
Burned plates	5,174	478
Laminated plates	565	50
Cases of defective riveting	3,225	610
Defective heads	1,204	166
Cases of leakage around tubes	13,015	1,789
Cases of defective tubes	9,691	2,508
Tubes too light	2,009	552
Leakage of joints	5,956	353
Water gauges defective	3,402	668
Blow-offs defective	4,436	1,288
Cases of deficiency of water	430	122
Safety valves overloaded	1,209	354
Safety valves defective	1,334	356
Pressure gages defective	8,145	469
Boilers without pressure gauges	369	369
Unclassified defects	9	4
	169,202	16,746
Condemned	653	

The neutralization or elimination of the impurities may be effected by one of the following methods:

1. Chemically.
   - Boiler compounds.
   - Purifying plants.
2. Mechanically.
   - Filters.
   - Blow-off.
   - Tube cleaners.
3. Thermally.
   - Feed-water heater.
   - Distillation.

The following chart ("Boiler Waters," W. W. Christie) outlines some of the troubles arising from feed water, their cause and means for preventing them.

Trouble.	Cause.	Remedy or Palliation.
Incrustation.	Sediment, mud, clay, etc.	Filtration. Blowing off.
	Readily soluble salts	Blowing off.
	Bicarbonate of magnesia, lime, iron	Heating feed and precipitate. Caustic soda. Lime. Magnesia.
	Organic matter	See below.
	Sulphate of lime	Sodium carbonate. Barium chloride.
Corrosion.	Organic matter	Precipitate with alum / Precipitate with ferric chloride — and filter
	Grease	Slaked lime / Carbonate of soda — and filter
	Chloride or sulphate of magnesium	Carbonate of soda.
	Sugar Acid	Alkali.
	Dissolved carbonic acid and oxygen	Slaked lime. Caustic soda. Heating.
	Electrolytic action	Zinc plates.
Priming.	Sewage	Precipitation with alum or ferric chloride and filter.
	Alkalies	Heating feed and precipitate.
	Carbonate of soda in large quantities	Barium chloride.

Feed water is considered *hard* when the mineral matter in solution curdles or precipitates soap. The constituents which cause this "hardness" are carbonates and sulphates of lime, magnesia, and iron. Hardness, due to the bicarbonates, which is reduced by boiling, is said to be "temporary" while that which is not removed in this way is said to be "permanent." Low hardness, to 200 parts of calcium carbonate per million, is conveniently determined by means of a standard soap solution. The latter may be obtained from chemical dealers. In determining the degree of hardness, 50 cubic centimeters of the water are introduced into a 200-cubic-centimeter bottle and alcoholic soap solution is added from a burette until a lather is obtained which covers the entire surface of the liquid and is permanent for five minutes. The degree of hardness is calculated by the use of Clark's table, thus:

CLARK'S TABLE OF HARDNESS.

Standard Soap Solution, Cubic Centimeters.	Parts of $CaCO_3$ Per Million.	Standard Soap Solution, Cubic Centimeters.	Parts of $CaCO_3$ Per Million.	Standard Soap Solution, cubic centimeters.	Parts of $CaCO_3$ Per Million.
0.7	0.0	6.0	74.0	12.0	164.0
1.0	5.0	7.0	89.0	13.0	180.0
2.0	19.0	8.0	103.0	14.0	196.0
3.0	32.0	9.0	118.0	15.0	212.0
4.0	46.0	10.0	133.0		
5.0	60.0	11.0	148.0		

For waters which are harder than 200 parts per million use a solution ten times the strength of the standard. This method does not indicate the amount of reagent to be used in neutralizing the calcium sulphate but gives an idea of the quantity of soda crystals required to soften the water as far as the $CaCO_3$ is concerned.

The most satisfactory way is to submit a sample of the feed water to a competent chemist for analysis and add the reagent recommended.

Complete Examination of Water for Boiler Purposes: Chem. Engr., Feb., 1910, p. 41; Mech. Engr., Aug. 16, 1910, p. 247, May 31, 1912, p. 692; Met. and Chem. Engng., Jan., 1910, p. 21; Jour. Frank. Inst., Vol. CLIX, p. 217.

Boiler Feed Water: Cassier's Mag., Oct., 1911, p. 561; June, 1910, p. 189, Jour. Indus. and Engng. Chem., May, 1911, p. 326.

Boiler Corrosion: Power, June 13, 1911, p. 910, July 11, 1911, p. 67; Boilermaker, Aug., 1912, p. 253; Prac. Engr., U. S., Sept. 1, 1912, p. 881.

Boiler Scale Prevention: Elec. Wld., Jan. 6, 1910, p. 46, July 1, 1909, p. 31; Power, Apr. 16, 1912, p. 560, May 17, 1910, p. 888.

274. Chemical Purification. — Chemical treatment of boiler feed water has been remarkably developed during recent years and a number of manufacturing concerns make this their sole business. The two most common systems of chemical treatment involve (1) boiler compounds and (2) purifying plants. In the former the necessary chemical action takes place inside the boiler and in the latter the water is purified before it enters the boiler. In either case the usual procedure is to submit for analysis a sample of the feed water and the resulting scale to a competent chemist who will specify the character and quantity of chemicals necessary to bring about the desired result.

275. Boiler Compound. — The object of treatment with boiler compounds is to neutralize the evil effects of the impurities in the feed water or to change them into others which are less objectionable and which are easily removed. When properly compounded and introduced into the boiler such preparations are of great benefit and practically overcome the deleterious effects, but when improperly used

they may produce even greater troubles than the impurities which they are expected to eliminate.

Boiler compounds may be divided into three classes:

1. Those converting the scale-forming elements into new substances which will not form a hard, resisting scale and which are readily removed by skimming, blowing off, or by tube cleaners. For example, feed water containing sulphates of lime and magnesia will form a dense, tenacious scale. If carbonate of soda be added in correct amount, the sulphates are converted into insoluble carbonates which are precipitated and form scale varying from a more or less porous, friable crust to a soft "mush" or mud. The resulting sulphate of soda remains in solution and does not form scale unless allowed to concentrate, and this is prevented by blowing off. An excess of soda is apt to cause foaming and at high temperatures is liable to attack the inside of gauge glasses. Bisodium and trisodium phosphate, sodium tannate, fluoride of sodium, sugar, etc., have all proved satisfactory, but as each case requires special treatment no detailed discussion is possible within the scope of this work and the reader is referred to the accompanying bibliography.

2. Those enveloping the newly precipitated scale-forming crystals with a surface which prevents them from cementing together. The ingredients used to bring about this result are starches, woody fibers, dextrine, slippery elm, and the like.

3. Those preventing the formation of hard scale by a solvent or "rotting" action, as kerosene and petroleum oils.

Boiler Compounds. — *Use of Compounds:* Eng. News, July 27, 1905, p. 112; Am. Mach., Dec. 7, 1899, p. 115, Oct. 26, 1899, p. 1014; Power, Aug., 1903; Eng. and Min. Jour., Aug. 12, 1905, p. 253; Elec. Wld., Oct. 7, 1909, p. 844.

276. Use of Kerosene and Petroleum Oils in Boiler Feed Water. — Kerosene oil and other refined petroleum oils are sometimes used with good effect in boilers to prevent scale from adhering. These oils are said to change the deposit of lime from a hard scale to a friable material which may be easily removed. They are ordinarily fed to the boiler with the feed water, drop by drop, through a sight feed apparatus similar to a cylinder oil lubricator. From extended experiments made on a 100-horse-power tubular boiler fed with water containing 6.5 grains of solid matter per gallon it was found that one quart of kerosene per day was sufficient to keep the boiler entirely free from scale. Prior to the introduction of the oil the water had a corrosive action upon some of the fittings attached to the boiler, but after the oil had been used for a few months it was found that the corrosive action had ceased. In another case 40 gallons of kerosene were used in 24 hours in a steamer

of about 3000 horse power. These boilers showed no incrustation but considerable corrosion. Evidently oil does not have the same effect or give the desired results in all cases. Kerosene used in moderate quantities will not cause foaming. Crude oil should never be used, as the heavy residue causes the formation of a tough, impervious scale productive of bagged sheets and collapsed flues.

Use of Kerosene in Boilers: Engr. U. S., Sept. 15, 1905, p. 634; Eng. News, May 24, 1890, p. 497; Power, Nov. 8, 1910, p. 1993; Trans. A.S.M.E., 9–247, 11–937; Locomotive, July, 1890, p. 97.

277. Use of Zinc in Boilers. — Zinc is often introduced into boilers to prevent corrosion. The theory is that a feeble but continuous current of hydrogen is generated over the whole extent of the iron by electrolytic action. The bubbles of hydrogen formed isolate the metallic surface from scale-forming substances. If there is but a little of the scale-forming element it is precipitated and reduced to mud; if there is considerable, coherent scale is produced which takes the form of the iron surface but does not adhere to it, being prevented from doing so by the intervening bubbles of hydrogen. Zinc is ordinarily suspended in the water space of the boiler in the shape of blocks, slabs, or as shavings in a perforated vessel. Electrical connection between the metallic surfaces is essential. Rolled zinc slabs $12 \times 6 \times \frac{1}{2}$ inches have found much favor in marine practice. Generally speaking one square inch of zinc surface is sufficient for every 50 pounds of water in the boiler, though the quantity placed in the boiler should vary with the hardness. The British Admiralty recommends the renewing of the zinc slabs whenever the decay has penetrated to a depth of $\frac{1}{4}$ inch below the surface. Zinc does not prevent corrosion or scale formation in all cases and may even aggravate the trouble.

Use of Zinc in Boilers: Prac. Engr., Dec., 1911, p. 835; Power, Oct. 18, 1910, p. 1874; Sept. 27, 1910, p. 1734.

278. Methods of Introducing Compounds. — Boiler compounds may be introduced into the boiler continuously or intermittently. Small quantities introduced continuously or at short intervals are more effective than large quantities at long intervals. Continuous feeding is ordinarily brought about by connecting the suction side of the feed pump with a reservoir containing the compound in solution, arranged similarly to an ordinary cylinder oil lubricator. In large plants an independent pump is often used to force the solution into the feed line. Intermittent feeding is brought about by temporarily connecting the suction of the feed pump with the reservoir containing the compound. The use of boiler compounds does not necessarily prevent scale from

forming in time, though it will reduce the evil to a minimum. In some instances where compounds are used it is found necessary to run a tube cleaner through the tubes at certain intervals, in others such a course has not been found necessary.

279. Weight of Compound Required. — The weight of compound introduced depends upon the nature of the reagents used and the character of the feed water, and ranges from a few ounces to several pounds per 100 gallons of feed water. For example, water containing 4 grains of calcium sulphate and 6 grains of magnesium sulphate per gallon will require 3.57 pounds of carbonate of soda per 1000 gallons of water for the reduction of the sulphates. The chemical reaction and analysis is as follows:

$$CaSO_4 + Na_2CO_3 = CaCO_3 + Na_2SO_4,$$
$$MgSO_4 + Na_2CO_3 = MgCO_3 + Na_2SO_4.$$

If x = grains of Na_2CO_3 necessary for the calcium,

$$CaSO_4 : Na_2CO_3 + 10H_2O = 4 : x.$$
$$40 + 32 + 4 \times 16 : 2 \times 23 + 12 + 3 \times 16 + 10\,(2 + 16) = 4 : x.$$
$$x = 8.41 \text{ grains.}$$

If y = grains of Na_2CO_3 necessary for the magnesium,

$$MgSO_4 : Na_2CO_3 + 10H_2O = 6 : y.$$
$$24 + 32 + 4 \times 16 : 2 \times 23 + 12 + 3 \times 16 + 10\,(2 + 16) = 6 : y.$$
$$y = 14.3.$$

The total weight of carbonate of soda per 1000 gallons is therefore

$$1000\,(14.3 + 8.41) = 22{,}710 \text{ grains}$$
$$= 3.24 \text{ pounds.}$$

This amount would effect the desired result if the chemical reaction is permitted to take place for some time, otherwise an excess of reagent is necessary. As a rule, however, one quarter of the theoretical quantity calculated is used in boiler feed practice.

280. Mechanical Purification. — Waters containing sand, mud, organic matter, and in fact all matter which is not in solution or in chemical combination with the water may be purified by mechanical filtration. Mud and sand may be eliminated by simply permitting the water to stand for some time in settling tanks. Suspended matter which will not gravitate to the bottom may be removed by filtering the water through coke, cloth, excelsior, or the like. Filters should be in duplicate for continuity of operation.

Vegetable and other organic impurities commonly float on the surface of the water when the boiler is making steam, and may be blown out through a "surface blow-out." (See paragraph 88.)

Precipitated matter may be ejected from the boiler by frequent blowing off before it has time to adhere and bake to a crust. This procedure is particularly essential when boiler compounds are used.

For description and use of mechanically operated tube cleaner see paragraph 92.

281. Thermal Purification. — (See also Live Steam Purifiers, paragraph 298.) The carbonates of lime and magnesia are held in solution in fresh water by an excess of carbon dioxide and are completely precipitated by boiling. At ordinary temperatures carbonate of lime is soluble in approximately 20,000 times its volume of water, at 212 degrees F. it is slightly soluble, and at 290 degrees it is insoluble. Sulphate of lime is much more soluble in cold than in hot water, and is completely precipitated at 290 degrees. (Revue de Mécanique, November, 1901, pp. 508, 743.) Thus it will be seen that a feed heater may be relied upon to remove part or all of the lime, depending upon the temperature to which the water is raised and the time in which the precipitation is permitted to take place.

Influence of Temperature and Concentration on the Saline Constituents of Boiler Water: Jour. Soc. Chem. Ind., Oct. 31, 1900, p. 885. *Solubility of Sulphate of Lime:* Rev. de Mécanique, Jan., 1901, p. 5, Nov., 1901, p. 508.

282. Purifying Plants. — The function of a purifying plant is the elimination of all impurities from the feed water before it enters the boiler. Purifying plants are continuous or intermittent in operation and are modified in a number of ways to meet different conditions.

A typical continuous system is illustrated in Fig. 343. The hard water enters the softener through the inlet pipe, is discharged into the raw water box, whence it passes over the water wheel, and thus generates the power necessary to maintain the reagents in constant agitation. From the water wheel the hard water passes into the top of the cone, where it meets the reagents delivered by the lift pipe and is thoroughly mixed with them. The reagents are dissolved in the mixing tank, located at the ground level, and by means of a steam, electric, or power pump are then elevated into the chemical tank above. One charge is sufficient to last ten hours or more. The reagents are apportioned to the amount of incoming raw water to the dividing box. (Inasmuch as the "head" over this stream varies directly with any fluctuation of the main hard water stream, the two streams are constantly maintained in the same proportion to each other.) In the dividing box this small stream is again divided by a slide which throws one part of the water back into the hard water stream and another part — which determines the rate of flow of the chemicals — into the regulating tank. As the level of water in the regulating tank rises, the float rises likewise and

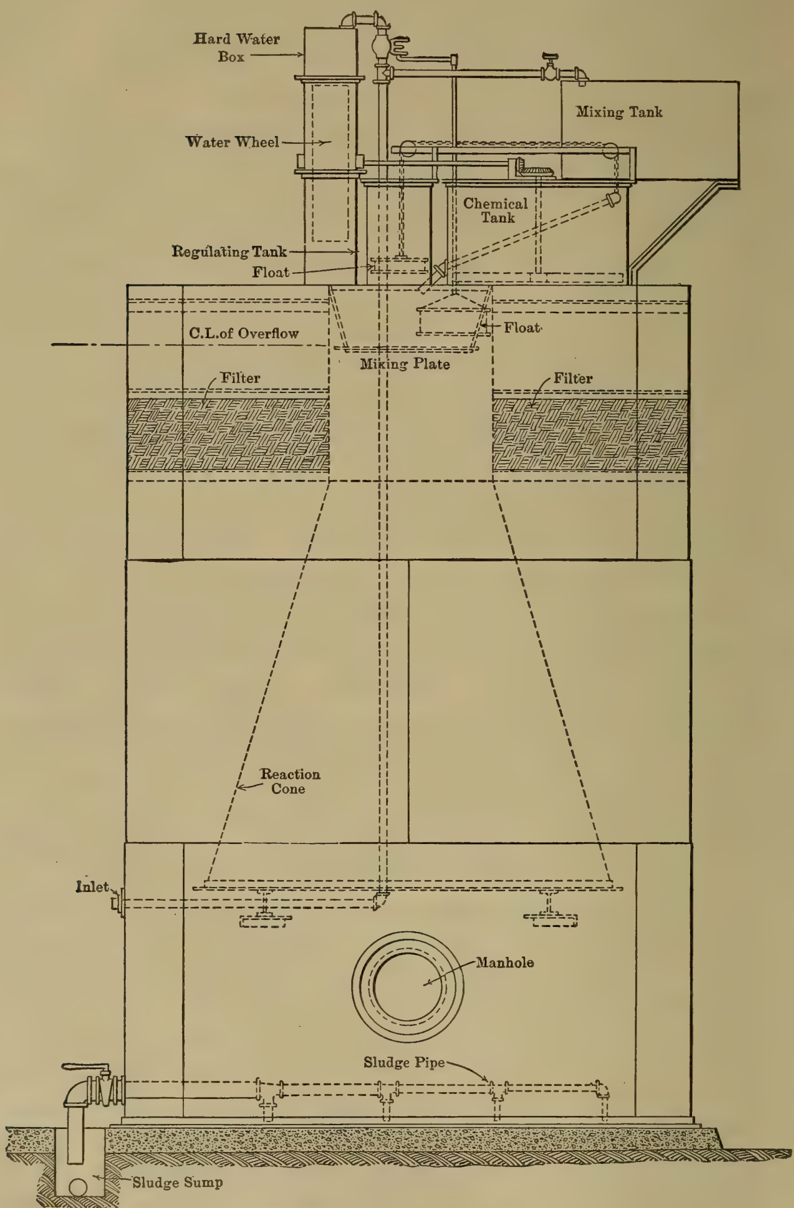


FIG. 343. Kennicott Type K Feed-water Purifier.

by means of a connecting chain lowers the mouth of the lift pipe in the reagent tank. Through this lift pipe the reagents flow into the top of the cone and intimately mix with the raw water. The reaction between the raw water and the reagents starts as soon as they meet, and as the mixture flows from the mixing plate into the reaction cone or downtake, the precipitation of the scale-forming and soap-destroying material commences to take place. Flowing at a constantly decreasing rate, owing to the constantly increasing diameter of the channel, the water passes to the bottom of the cone, turns and flows upward still at a constantly decreasing rate, the precipitate falling away from it as it moves. Finally the water passes through a filter which removes any slight trace of precipitate that remains; and it then is discharged from the top of the softener. The precipitate, which consists of the impurities of the raw water and the softening chemicals in chemical union, falls to the bottom of the main tank and is from time to time discharged therefrom through a sludge valve. An electric indicator is provided which rings a bell one-half hour before a new supply of reagents is needed and thus notifies the attendant of the fact. The lift pipe is a tube, flexible for a portion of its length, through which the chemicals leave the chemical tank. By means of the regulating device the mouth of this tube is maintained at a constant depth of immersion in the surface of the dissolved reagents.

In the Scaife system for water purification feed water first enters the heater, where it attains a temperature of from 200 to 210 degrees F. As a portion of the free CO_2 is driven off by the heat the carbonates of lime and magnesia are precipitated and are deposited in removable pans inside the heater. On its way the heated water is forced by the boiler feed pump into a large precipitating tank, where the necessary chemicals are introduced by two small pumps. These pumps take the solution of chemicals from the solution tanks which hold a sufficient quantity to operate the plant from eight to twelve hours. The precipitating tank is so constructed as to cause intimate and thorough mixing of the chemicals with the water. Thus the acids are neutralized, and the scale-forming substances are precipitated by being changed to insoluble substances which sink to the bottom of the precipitating tank whence they are readily removed. Some of the lighter substances remaining in suspension are carried along with the water as it passes into the filters, which effectively remove all suspended matter. This system is continuous in operation, and purification is accomplished without appreciably retarding the onward flow of feed water. Fig. 344 shows a modification of the system. The chemicals are pumped from the "chemical tank" into the "solution tanks," where the feed water and

chemical solution are thoroughly mixed. The treated water is taken from these tanks and pumped into the "precipitating tanks" where a large portion of the scale-forming element is precipitated. From the precipitating tanks the water is forced through a series of filters to the boiler.

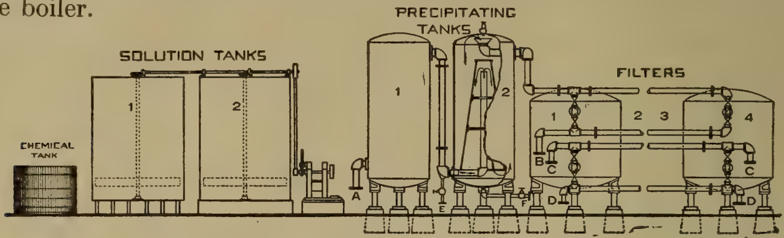


FIG. 344. General Arrangement of Scaife System of Feed-water Purification.

Fig. 345 illustrates the We-Fu-Go system of water purification. In this installation the water supply first enters the settling or treating tanks into which the chemicals are fed. A thorough mixture is effected by the use of the two armed paddles located near the bottom of the tanks. From the treating tanks the water flows by gravity into the filters, which remove all remaining impure solid matter which does not settle to the bottom of the treating tank. The pipes conducting the water from the settling tanks to the filter are fitted with a flexible joint and float so that the outlets are near the surface at all times, rising and falling with the water level. From the filters the purified water gravitates into the clear water storage reservoir, from which it is pumped into an open heater and thence to the boiler. This system is intermittent in operation, and in order to provide sufficient time for thorough chemical treatment of large quantities, two or more settling tanks are employed. Both the We-Fu-Go and Scaife systems are modified in a number of ways to meet different conditions.

Fig. 346 shows the general arrangement of the Anderson system for preventing corrosion in condensers and removing oil from condensed steam. The method consists in injecting into the exhaust steam as it passes from the preheater to the condenser a solution containing a coagulant which changes the emulsion of the cylinder oil to a flaky condition so that it may be separated by settling, flotation, or filtering. The air pump delivers the water to the settling tank *F*, whence it is taken to the open gravity filters *G*, *G*, of a superficial area proportional to the amount of water to be passed and containing a filter bed of four feet of crushed quartz. This will run about four days without any marked difference in efficiency, after which time the bed is stirred to a depth of two feet by mechanical agitators and flushed with clean water, by which all impurities are carried to the sewer. The solution is pre-

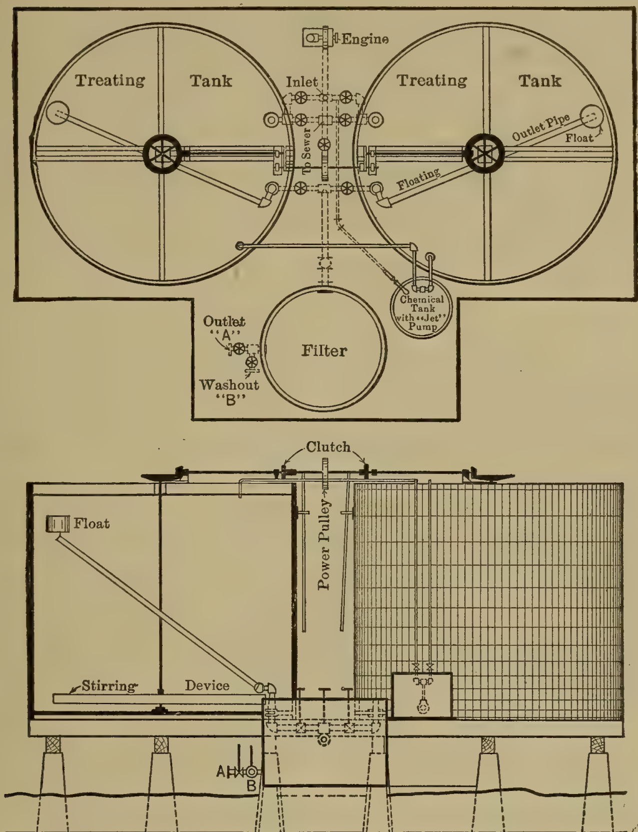


FIG. 345. General Arrangement of We-Fu-Go System of Feed-water Purification.

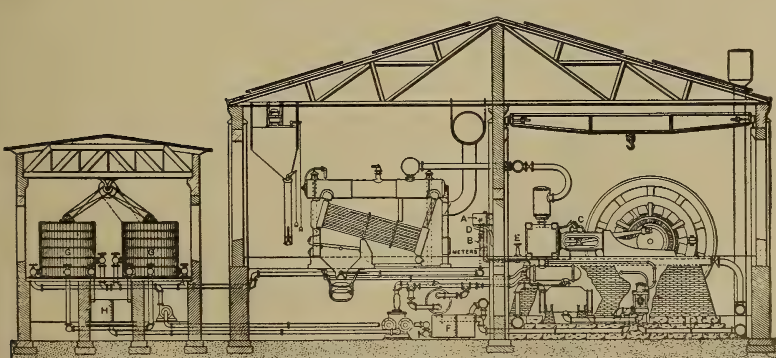


FIG. 346. Anderson System for Preventing Corrosion in Condensers.

pared in tank A, in which the water level is preserved by a ball float and into which filtered water is admitted through pipe B, while the substance with which the water is treated is pumped in through the pipe D by a small pump operated from the main engine. The flow to the "rose head" above the condenser is controlled by the valve E, and a meter in this pipe records the amount being fed. The water ordinarily required for "make up" is sufficient to carry in the solution. There is very little loss of water, and the rapid corrosion of the condenser tubes, which has been so great an obstacle to the successful use of surface condensers, is much reduced. The chemicals used perform a twofold duty, viz., to neutralize the water and make it chemically inactive and to coagulate the oily matter contained in the steam so that mechanical filtration is possible. (Power, June, 1903, p. 304.)

Water-softening plants cost from $4 to $5 per horse power for plants of 1000 horse power and less, from $3 to $4 for plants of 1000 to 2000 horse power, and as low as $1.50 for plants of 5000 horse power or more. The depreciation of wooden tanks is as high as 15 per cent a year, while that of steel tanks should not be greater than 5 per cent. Unless wooden tanks are considerably cheaper than steel tanks they are not a good investment. The cost of water purification varies from a fraction of a cent to 2 cents per 1000 gallons, depending upon the size of the plant and the quantity and character of the impurities. (American Electrician, March, 1905, p. 125.)

Water Softening and Treatment for Power Plant Purposes: Chem. Engr., Jan., 1910, p. 5; Eng. News, June 6, 1912, p. 1087; Ry. Age Gazette, Aug. 16, 1912, p. 288; Ry. Master Mechanic, May, 1910, p. 153; Power, May 28, 1912, p. 780; Apr. 18, 1911, p. 598; Prac. Engr., U. S., Mar., 1910.

283. Economy of Preheating Feed Water. — Although a feed-water heater acts to some extent as a purifier its primary function is that of heating the feed water. Generally speaking, for every 10 degrees that the feed water is heated there is a gain in heat of 1 per cent and a corresponding saving of coal, if the heat which warms the feed water would otherwise be wasted. Again, the smaller the difference in temperature between the steam and the feed water the less will be the strain on the boiler shell due to unequal expansion and contraction, an item of no small consequence.

If H represents the heat content of the steam above 32 degrees F., t_0 the temperature of the cold water, and t the temperature of the water leaving the heater, then S, the per cent gain in heat due to heating the feed water, may be expressed

$$S = 100 \frac{(t - t_0)}{H - (t_0 - 32)}. \qquad (196)$$

The expression is not theoretically correct, since it assumes a constant value of unity for the specific heat, whereas the specific heat varies with the temperature. The variation is so slight, however, that it may be neglected for all practical purposes.

Example: Steam pressure 100 pounds gauge; temperature of water entering heater 80 degrees F.; temperature of water leaving heater 210 degrees F. Required, saving due to heating the feed water.

Here H (from steam tables) is 1188, t_0 = 80, t = 210.

$$S = 100 \frac{(210 - 80)}{1188 - (80 - 32)}$$
$$= 11.4 \text{ per cent.}$$

This formula gives the *thermal* saving only, and the first cost of the heater, interest, depreciation, attendance, and repairs must be taken into consideration before the *net* saving measured in dollars and cents is ascertained. In the average installation the net saving is a substantial one.

Table 91 based upon formula (196) may be used in determining the percentages of saving due to the increase in feed-water temperature.

TABLE 91.

PERCENTAGE OF SAVING FOR EACH DEGREE OF INCREASE IN TEMPERATURE OF FEED WATER.

(Based on Marks & Davis Steam Tables.)

Initial Temp. of Feed.	Boiler Pressure Above Atmosphere.										
	0	20	40	60	80	100	120	140	160	180	200
32	.0869	.0857	.0851	.0846	.0843	.0841	.0839	.0837	.0835	.0834	.0834
40	.0875	.0863	.0856	.0853	.0849	.0846	.0845	.0843	.0841	.0840	.0839
50	.0883	.0871	.0864	.0859	.0856	.0853	.0852	.0850	.0848	.0847	.0846
60	.0891	.0878	.0871	.0867	.0864	.0861	.0859	.0857	.0855	.0854	.0853
70	.0899	.0886	.0879	.0874	.0871	.0868	.0867	.0865	.0863	.0862	.0861
80	.0907	.0894	.0887	.0882	.0878	.0876	.0874	.0872	.0871	.0870	.0869
90	.0915	.0902	.0895	.0890	.0887	.0884	.0882	.0880	.0878	.0877	.0876
100	.0924	.0910	.0903	.0898	.0895	.0892	.0890	.0888	.0886	.0885	.0884
110	.0932	.0919	.0911	.0906	.0903	.0900	.0898	.0896	.0894	.0893	.0892
120	.0941	.0927	.0919	.0915	.0911	.0908	.0906	.0904	.0902	.0901	.0900
130	.0950	.0936	.0928	.0923	.0919	.0916	.0915	.0912	.0911	.0910	.0909
140	.0959	.0945	.0937	.0931	.0928	.0925	.0923	.0921	.0919	.0918	.0917
150	.0969	.0954	.0946	.0940	.0987	.0933	.0931	0930	.0928	.0927	.0926
160	.0978	.0963	.0955	.0948	.0946	.0942	.0940	.0938	.0936	.0935	.0934
170	.0988	.0972	.0964	.0958	.0955	.0951	.0948	.0947	.0945	.0944	.0943
180	.0998	.0982	.0973	.0968	.0964	.0960	.0958	.0956	.0954	.0953	.0952
190	.1008	.0992	.0983	.0977	.0973	.0969	.0968	.0965	.0964	.0963	.0962
200	.1018	.1002	.0993	.0987	.0983	.0978	.0977	.0974	.0973	.0972	.0971
210	.1029	.1012	.1003	.0997	.0993	.0989	.0987	.0984	.0983	.0982	.0981
220	......	.1022	.1013	.1007	.1003	.0999	.0997	.0994	.0992	.0991	.0990
230	......	.1032	.1023	.1017	.1013	.1009	.1007	.1004	.1002	.1001	.1000
240	......	.1043	.1034	.1027	.1023	.1019	.1017	.1014	.1012	.1011	.1010
250	......	.1054	.1044	.1008	.1034	.1029	.1027	.1024	.1022	.1021	.1020

Multiply the factor in the table corresponding to any given initial temperature of feed water and boiler pressure by the total rise in feed-water temperature; the product will be the percentage of saving.

Feed-water Heating. — Power, June 25, 1912; Eng. News, Sept. 9, 1909, p. 284; Elec. Wld., March 2, 1911, p. 551; Mech. Engr., Nov. 5, 1909, p. 588; Engr. U. S., Jan. 1, 1906, p. 8, Aug. 15, 1904, p. 15; St. Ry. Jour., July 22, 1905, p. 145; Am. Elecn., Dec., 1904, p. 570; Am. Elecn., Nov., 1904; Engr., Lond., July 28, 1905.

284. Classification of Feed-water Heaters. — Feed-water heaters may be classified according to the *source* of heat, as

1. *Exhaust steam,* in which the heat is received from the exhaust of engines, pumps, etc.

2. *Flue gas,* in which the waste chimney gases are the source of the heat.

3. *Live steam purifiers,* or those using steam at boiler pressures; or according to the method of heat *transmission,* as

1. *Open heaters,* in which the steam and feed water mingle and the steam in condensing gives up its heat directly to the water.

2. *Closed heaters,* in which the steam and water are in separate chambers and the steam gives up its heat to the water by conduction.

Heaters may also be classified according to the pressure of the heating steam, as

1. *Vacuum* or *primary,* in which the pressure is less than atmospheric and applies particularly to heaters utilizing the exhaust of condensing engines. These are always of the closed type. Open heaters in which the pressure is less than atmospheric are not usually classed as vacuum heaters.

2. *Atmospheric* or *secondary,* in which the pressure is atmospheric or, literally, that corresponding to the back pressure on the engines and pumps.

3. *Pressure,* in which the pressure corresponds to that in the boiler and in which the heat is used primarily for purifying purposes.

CLASSIFICATION OF A FEW TYPICAL HEATERS.

Exhaust steam	Open	Atmospheric	Cochrane Hoppes Stillwell Webster	
	Closed	Atmospheric Vacuum or pressure	Wainwright Wheeler	Water Tube
			Otis Berryman	Steam Tube
Flue Gas			Green American Sturtevant	
Live Steam	Open	Pressure	Hoppes Baragwanath	

Heaters may be still further classified as

1. *Induced,* in which only such steam is admitted as is induced by its condensation. That is, the feed water condenses the steam. This creates a partial vacuum which draws in more steam.

2. *Through,* in which all the steam is forced through the heater irrespective of condensation.

285. Open Heaters. — Fig. 347 gives a sectional view of a Cochrane special feed heater and receiver and is a typical example of an open

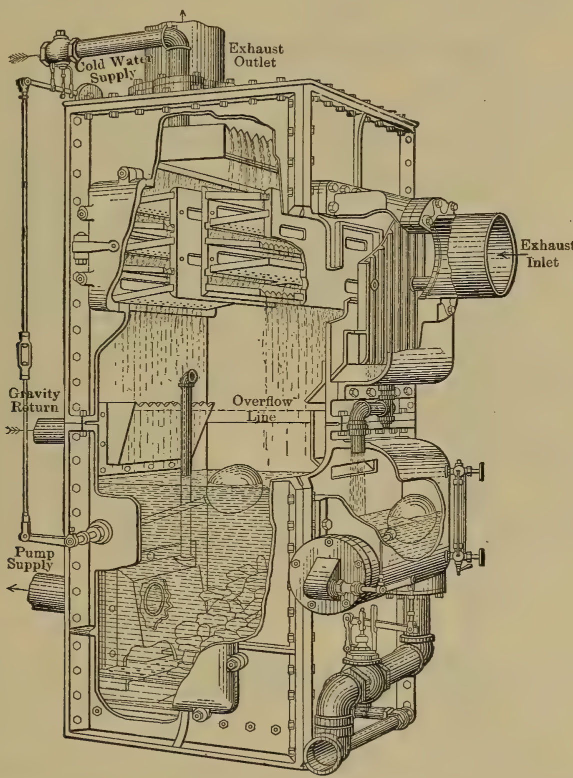


FIG. 347.

heater. Exhaust steam enters the heater through a fluted oil separator as indicated, and passes out at the top, while the oily drips are automatically drained to waste by a suitable ventilated float. The feed water enters through an automatic valve and is distributed over a series of copper trays so arranged and constructed that the water is forced to fall in a finely divided stream before reaching the reservoir in the bottom. The steam coming in contact with the water particles gives up latent heat and condenses. Much of the scale-forming element

is deposited on the surface of the trays, from which it is readily removed. The suspended matter is eliminated by a coke filter in the bottom of the chamber, and the floating impurities are decanted by a skimmer or overflow weir. The particular heater shown in the illustration is especially designed for use in a steam-heating plant; i.e., besides performing all the functions of an open heater, it provides for the reception and heating of the condensation returned to it from the heating system.

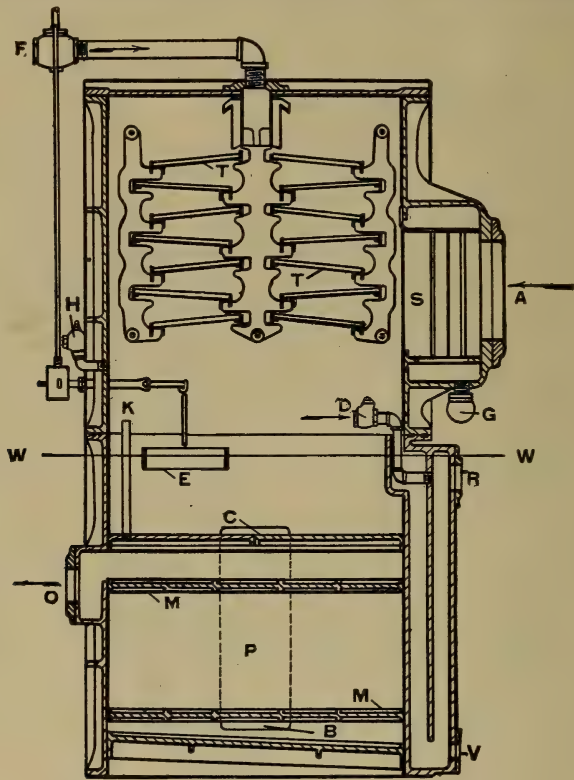


FIG. 348. Section Through Webster Heater.

Fig. 348 gives a sectional view of a Webster "star vacuum" heater. Water enters the heater through balanced valve *F*, which is controlled by float *E*, and is deflected over a series of perforated copper trays *T*, *T*. Exhaust steam enters at *A*, passes through oil filter *S*, and, mingling with the finely divided streams of water, gives up its latent heat and is condensed. Only so much steam enters the heater as is condensed by the feed water. The condensed steam and feed water fall to the bottom of the upper chamber, maintaining a practically constant level *WW*. From this upper or heater chamber the water gravitates to the settling

chamber at the bottom, through down-cast pipe *CB*. From the settling chamber the water rises through perforated screen *M* and filtering material *P* to the outlet *O*. A large portion of the scale-forming element is precipitated on the trays or collects in the settling chamber at the bottom.

Fig. 349 shows a section through a Hoppes open heater, illustrating the "pan" type. Exhaust steam enters at *H*, passes through oil filter *O*, and completely surrounds pans *T*, *T*. The feed water enters at *B*, and the rate of flow is regulated by valve *F*, which is controlled by a

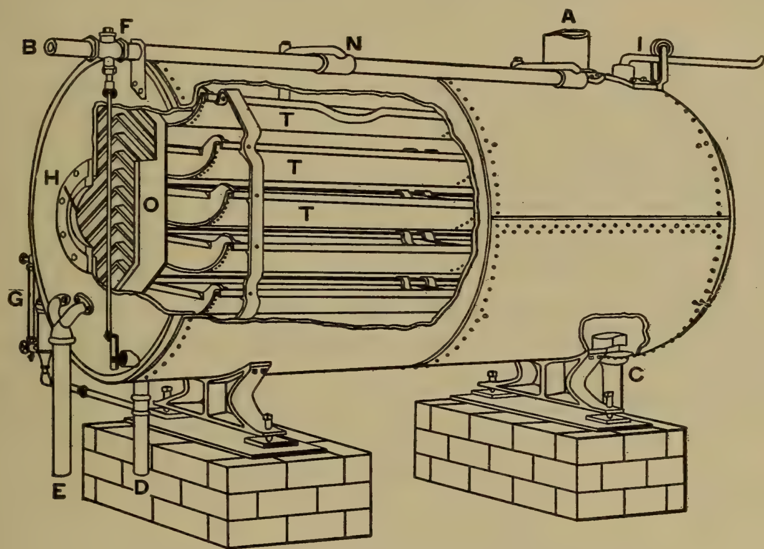


FIG. 349. Hoppes Horizontal Feed-water Heater.

suitable float in the lower part of the chamber. The water in flowing over the sides and bottoms of the pans comes in direct contact with the steam.

286. Combined Open Heater and Chemical Purifier. — Combined feed-water heaters and chemical purifiers are finding increased favor with engineers in many districts where the feed water is particularly bad. A description of the Webster combination will be found in Part II of the general catalogue issued by the Warren Webster Company, Camden, N. J. A description of the Cochrane-Sorge combined heater and chemical purifier will be found in the heater catalogue issued by the Harrison Safety Boiler Works, Philadelphia, Pa.

287. Temperatures in Open Heaters.—The temperature to which feed water is raised in an open heater may be determined as follows:

Let H represent the heat content of the steam entering the heater,
t_0 the temperature of the water entering heater,
t the temperature of the water leaving heater, and
S the ratio of exhaust steam to the feed water, by weight.

Then, allowing a loss of 10 per cent due to radiation, etc., $0.9\,S\,(H - t + 32)$ will be the B.t.u. given up by the exhaust steam to each pound of feed water, and $(t - t_o)$ will be the B.t.u. absorbed by each pound of water.

Therefore $0.9\,S\,(H - t + 32) = t - t_o$, from which

$$t = \frac{t_0 + 0.9\,S\,(H + 32)}{1 + 0.9\,S}. \tag{197}$$

If more steam passes through the heater than can be condensed by the feed water, then this equation gives t a fictitious value; in other words, t can never be greater than the temperature of the exhaust steam.

Substituting $t = 212$, the maximum obtainable temperature with exhaust steam at atmospheric pressure, and solving for S, we find that only 17 per cent of the main engine exhaust is necessary to heat the feed water to a maximum. t_o is assumed to be 60 degrees F.

Table 92 has been determined from this equation and gives the final temperatures obtainable in open heaters for various conditions of operation.

TABLE 92.

FINAL FEED-WATER TEMPERATURES. OPEN HEATER.

(Temperature of steam, 212 degrees F.)

Per Cent of Total Steam Used by Auxiliaries.	Initial Temperature of Feed Water, Degrees F.									
	40	50	60	70	80	90	100	110	120	130
2	60.1	69.9	79.7	89.5	94.4	109.2	119.0	128.8	138.7	148.5
3	69.9	79.6	89.3	90.1	108.8	118.6	128.3	138.0	147.8	157.5
4	79.5	89.1	98.8	108.5	118.1	127.8	137.4	147.1	156.7	166.4
5	89.0	98.5	108.1	117.7	127.2	136.8	146.4	155.9	165.5	175.1
6	98.3	107.7	117.2	126.7	136.2	145.7	155.2	164.7	174.2	183.6
7	107.4	116.8	126.2	135.6	145.0	154.4	163.8	173.2	182.5	192.1
8	116.4	125.7	135.0	144.4	153.7	163.0	172.4	181.8	191.0	200.3
9	125.2	134.5	143.7	153.0	162.2	171.5	180.7	190.0	199.2	208.5
10	133.3	143.1	152.3	161.4	170.6	179.8	189.0	198.1	207.3	212.0
11	142.5	151.6	160.7	169.7	178.9	188.2	197.0	206.2	212.0*	212.0*
12	150.9	159.9	168.9	177.9	187.0	196.0	205.0	212.0*	212.0*	212.0*

* All of the steam not condensed.

Example: A power plant has 1200 i.h.p. of engines using 20 pounds of steam per i.h.p. hour. Auxiliaries use equivalent of 10 per cent of main engine steam. Pressure in heater 0 pounds gauge, temperature

of hot-well supply 110 degrees F. Required temperature of feed water leaving heater.

Here $H = 1150$ (from steam tables), $t_0 = 110$, $S = 0.10$.

Substituting these values in (105),

$$0.9 \times 0.10\,(1150 - t + 32) = t - 110.$$
$$t = 198 \text{ degrees F.}$$

288. Pan Surface Required in Open Feed-water Heaters. — Pan or tray surface required varies according to the quality of the water with regard to both scale-making material and mud, and may be approximated by the formula

$$\text{Pan surface, sq. ft.} = \frac{\text{Lb. of water heated per hr.} \times \text{horse power}}{c}. \quad (198)$$

	Vertical Type.	Horizontal Type.
For very muddy water, c	118	110
Slightly muddy water, c	166	155
For clean water, c	500	400

289. Size of Shell, Open Heaters. — General proportions of open heaters vary considerably on account of the different arrangements of pans or trays, filter and oil-extracting devices. A fair idea of the size of shell required may be obtained by the formulas

$$\text{Area of shell} = \frac{\text{Horse power}}{a \times \text{length in feet}}, \quad (199)$$

$$\text{Length of shell} = \frac{\text{Horse power}}{a \times \text{area in square feet}}, \quad (200)$$

$a = 2.15$ very muddy water,
$a = 6$ for slightly muddy water,
$a = 8$ for clean water.

The horse power in this case is obtained by dividing the weight of water heated per hour by the steam consumption of the engine per horse power per hour.

Pans containing 2.5 square feet and less are usually made round, and larger sizes rectangular in plan. When circumstances will permit it is better to have not more than six pans in any one tier, since it is advisable to proportion the pans so as to obtain as low a velocity over each as practicable.

Distance between trays or pans is seldom less than one-tenth the width for rectangular and one-fourth the diameter for round pans.

Volume of storage and settling chamber in horizontal heaters varies from 0.25 for good quality of water to 0.4 of the volume of the shell for muddy water, 0.33 being about the average. In the vertical type the settling chamber represents respectively 0.4 and 0.6 the volume of the shell with clear and muddy water. Filters occupy from 10 to 15 per cent of the volume of the shell in the horizontal type and from 15 to 20 per cent in the vertical type, the smaller percentage corresponding to clear water and the larger to muddy water or water containing a considerable quantity of impurities.

Open Heaters: Cassier's Mag., Aug., 1903, p. 33; Engr. U. S., Jan. 1, 1906, pp. 17, 78; St. Ry. Jour., Feb. 4, 1905, p. 227; Elec. Wld., Apr. 27, 1911, p. 1051.

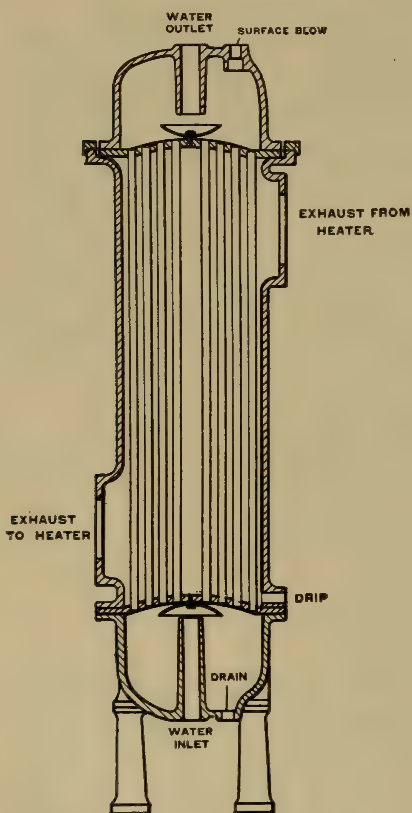


Fig. 350. Goubert Single-flow Closed Heater.

290. Classification of Closed Heaters. — Closed heaters may be grouped into two classes:

1. Water tube, Fig. 350, and
2. Steam tube, Fig. 354.

Closed heaters, both water tube and steam tube may operate with

1. *Parallel currents*, where the water and steam flow in the same direction, Fig. 353, or with

2. *Counter currents*, where the water and steam flow in opposite directions, Fig. 352.

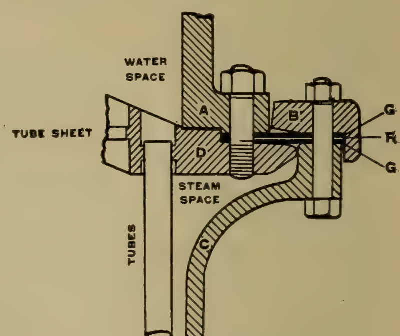


Fig. 351. Details of Expansion Joint, Goubert Heater.

Water-tube heaters may be still further classified as

1. *Single-flow*, in which the water flows through the heaters in one direction only, Fig. 350.

2. *Multi-flow,* in which the water flows back and forth a number of times, as in Fig. 352.

3. *Coil heater,* in which the water flows through one or more coils, as in Fig. 353.

291. Water-tube, Closed Heaters. — Fig. 350 shows a section through a feed-water heater of the single-flow straight-tube type. The tubes

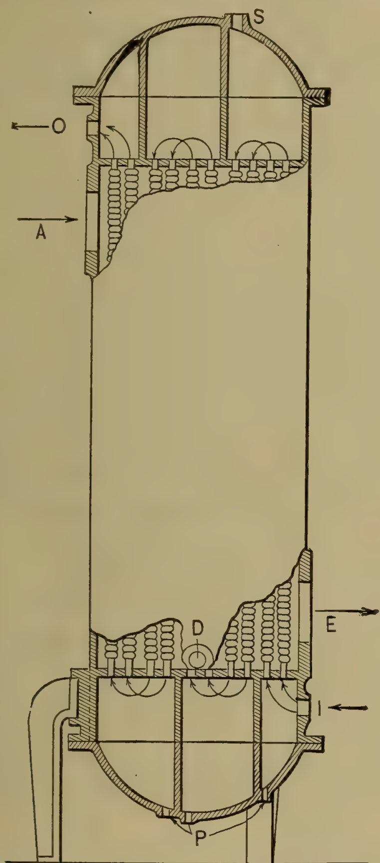


Fig. 352. Wainwright Multi-flow Closed Heater.

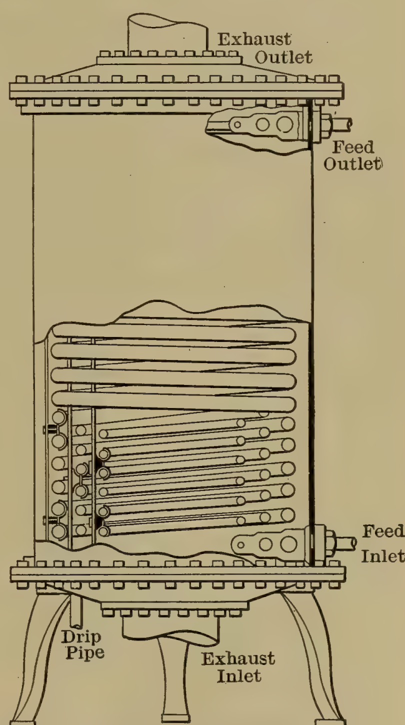


Fig. 353. Typical Coil Heater.

are of plain brass and the shell of cast iron. The tubes are expanded into the tube sheets by a roller expander. To provide for expansion the upper tube sheet and water chamber are secured to the main shell by means of a special expansion joint the details of which are shown in Fig. 351. *R* is a ring or gasket of soft annealed copper and *G*, *G* two gaskets of special packing with brass wire cloth insertion. These

gaskets form a flexible expansion joint between C and tube sheet D, so that the whole upper chamber, which is carried solely by the tubes, is free to move up and down as the tubes expand or contract under varying temperatures.

Fig. 352 shows a section through a Wainwright heater, illustrating the multi-flow water-tube type. The body of the heater is of cast iron, the tubes of corrugated copper. The water passes through the tubes and the steam surrounds them. The feed water and exhaust steam do not mingle, and hence the oil in the exhaust does not contaminate the water. The water chambers are divided into several compartments, as shown in the illustration, and the partitions are so arranged that the flow of feed water is directed back and forth through the various groups of tubes in succession. This arrangement gives a higher velocity of flow than the non-return type of heater, and therefore increases the rate of heat absorption. The mud and impurities settle at the bottom and are discharged through the mud blow-off. Such impurities as rise to the surface are removed by the surface blow-off. The tubes are corrugated to allow for expansion and at the same time to increase the transmission of heat. Referring to Fig. 352: Exhaust steam enters at A and leaves at E, and the portion which is condensed is drawn off at D. Feed water enters at I and is discharged at O. P, P are mud blow-offs and S is an opening for a safety valve. Fig. 356 gives results of tests showing the relative efficiencies of plain and corrugated tubes for various velocities.

Fig. 353 shows a partial section through a Harrisburg feed-water heater. This apparatus is a typical example of the coiled-tube heater. Three sets of concentric copper coils are brazed to gun-metal manifolds and supported by clamp stays as indicated in the illustration. Feed water enters the heater at the bottom manifold and passes through the coils to the feed outlet. The exhaust steam enters the heater at the bottom and surrounds the coils in its passage to the outlet at the top. The coils are designed to withstand a pressure of 600 pounds per square inch.

292. Steam-tube, Closed Heaters.—Fig. 354 shows a section through an Otis heater, illustrating the steam-tube type. Here the exhaust steam passes *through* the tubes which are surrounded by the feed water. The exhaust steam enters at A, and passes down one section of tubes into the enlarged space of the water and oil separator O, in which the condensation and oil are deposited. From this chamber the steam passes up through the other section of tubes to outlet C, thus passing twice through the entire length of the heater. The water enters at E and is discharged at G. R is the blow-off opening. The tubes are of

seamless brass and are curved to allow for expansion. Condensed steam is withdrawn at P.

Fig. 355 shows a partial section through a Baragwanath steam jacketed steam-tube heater. Exhaust steam enters at A, passes up through the tubes, returns down annular space E between the inner

FIG. 354. Otis Steam-tube Feed-water Heater.

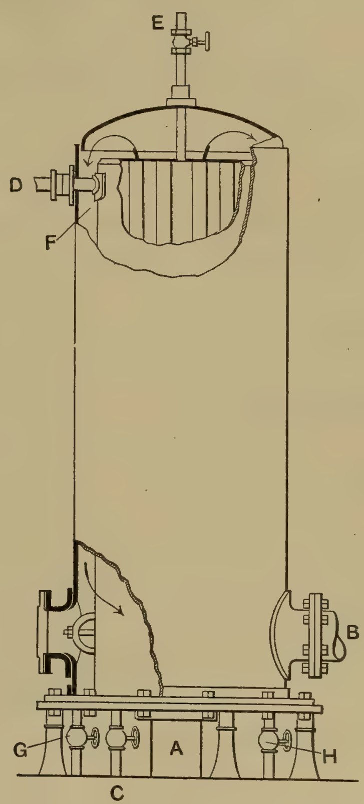


FIG. 355. Baragwanath Steam-jacketed Feed-water Heater.

shell and jacket, and passes out at B. Feed water enters at C and leaves at D. E is the scum blow-off, G the heater drain, and H the jacket drain.

293. Heating Surface, Closed Heaters. — It is generally assumed that the transfer of heat between two bodies is directly proportional to the difference in temperature between them.

Let t_0 = temperature of the water entering the heater;
t_2 = temperature of the water leaving the heater;
t_s = temperature of the exhaust steam;
A = square feet of transmitting surface;
t = temperature of a unit of water t' seconds after entering the heater;
h = B.t.u. absorbed per square foot per second per degree difference in temperature between the steam temperature t_s and the water temperature t;
t' = time in seconds;
w = number of pounds of feed water per second;

Then $\frac{A}{w}$ = square feet of surface brought in contact with one pound of water per second;

and dt, the rate at which the temperature of the water is increasing at this instant, will be

$$dt = \frac{hA}{w}(t_s - t)\,dt'. \tag{201}$$

$$\frac{dt}{t_s - t} = \frac{hA}{w}dt'. \tag{202}$$

Integrating,

$$\int \frac{dt}{t_s - t} = \frac{hA}{w}\int dt'. \tag{203}$$

$$\int_{t_0}^{t_2} \frac{dt}{t_s - t} = \frac{hA}{w}\int_0^{t'} dt'. \tag{204}$$

$$\log_e \frac{t_s - t_0}{t_s - t_2} = \frac{hAt'}{w}. \tag{205}$$

Let W = number of pounds of feed water heated per hour.
U = B.t.u. transmitted to the feed water per square foot of surface per hour per degree difference in temperature.

Then (205) may be written

$$\log_e \frac{t_s - t_0}{t_s - t_2} = \frac{AU}{W}, \tag{206}$$

from which

$$A = \frac{W}{U}\log_e \frac{t_s - t_0}{t_s - t_2}. \tag{207}$$

Knowing the weight of water to be heated, the temperature of the steam, the desired temperature of the feed water, and the coefficient of heat transmission, U, this equation enables one to determine the area of heating surface required for the given conditions. Since the extent of heating surface increases rapidly as t_2 approaches t_s, and becomes

infinity for $t_2 = t_s$, it is desirable to limit t_2 to some practical figure. An average maximum for $t_2 = t_s - 4$.

Table 93 has been calculated from this formula and gives the square feet of heating surface necessary to heat 1000 pounds of water per hour for different ranges in temperature.

Mean Temperature Difference.

If we let d = average temperature difference between the steam and feed water, then

$$AUd = \text{heat given out by the steam per hour,}$$

$$W(t_2 - t_0) = \text{heat absorbed by the feed water per hour,}$$

$$AUd = W(t_2 - t_0), \tag{208}$$

$$d = \frac{W(t_2 - t_0)}{AU}. \tag{209}$$

From (206),

$$\frac{AU}{W} = \log_e \frac{t_s - t_0}{t_s - t_2}.$$

Therefore

$$d = \frac{t_2 - t_0}{\log_e \dfrac{t_s - t_0}{t_s - t_2}}. \tag{210}$$

Equation 210 may be expressed

$$d = \frac{t - t'}{\log_e \dfrac{t}{t'}}, \tag{211}$$

in which t is the original temperature difference and t' the final temperature difference of the two fluids. Equation 211 is applicable to all conditions of parallel and counterflow.

Table 94 has been calculated from formula (210) and gives the mean temperature difference for various conditions of operation.

The arithmetic mean temperature difference d_1 may be taken with safety for the average heater problem and has the advantage of simplicity.

$$d_1 = t_s - \frac{t_0 + t_2}{2}. \tag{212}$$

Closed heaters are sometimes rated on the basis of $\frac{1}{3}$ square foot of heating surface per horse power, i.e., a heater with 500 square feet of heating surface would be rated at 1500 horse power.

294. Heat Transmission in Closed Heaters. — An inspection of the curves in Figs. 320 and 356 show that the absorption of heat per square foot of surface per degree difference in temperature varies with the

TABLE 93.

SQUARE FEET OF HEATING SURFACE REQUIRED TO HEAT 1000 POUNDS OF WATER PER HOUR.
$U = 350.$

Vacuum Heaters between Engine and Condenser.

Initial Temperature of Feed Water, t_0.	24″ Vacuum. Temperature 141° F. Final Temperature of Feed Water.						25″ Vacuum. $t_s = 134°$ F. Final Temperature of Feed Water.			
	105	110	115	120	125	130	110	115	120	125
40	2.93	3.36	3.86	4.50	5.22	6.29	3.93	4.65	5.58	7.01
50	2.64	3.29	3.57	4.15	4.93	6.01	3.65	4.36	5.28	6.65
60	2.29	2.93	3.22	3.86	4.58	5.65	3.29	4.01	4.93	6.29
70	1.93	2.50	2.86	3.43	4.22	5.29	3.07	3.58	4.57	5.86
80	1.50	2.07	2.43	3.01	3.72	4.86	2.36	3.07	4.01	5.36

t_0.	26″ Vac. $t_s = 125°$. Final Temperature of the Feed Water.				27″ Vac. $t_s = 114°$.			28″ Vac. $t_s = 100°$.		
	105	110	115	120	90	100	105	70	80	90
40	4.43	4.93	6.07	8.18	3.22	4.72	6.01	1.93	3.14	5.08
50	3.92	4.57	5.72	7.73	2.79	4.36	5.58	1.43	2.57	4.57
60	3.36	4.15	5.36	7.51	2.29	3.86	5.08	.78	1.93	3.93
70	2.86	3.65	4.79	6.86	1.71	3.22	4.50	......	1.14	3.14
80	2.29	3.07	4.28	6.28	.86	2.21	3.79	......	......	1.93

Atmospheric Heaters.

Initial Temperature of Feed Water, t_0.	Atmospheric Pressure. Temp. 212° F. Final Temperature of Feed Water.					
	192	196	200	204	208	210
40	6.01	6.65	7.58	8.73	10.72	12.74
50	5.94	6.58	7.44	8.58	10.51	12.51
60	5.79	6.44	7.15	8.36	10.38	12.30
70	5.58	6.22	7.01	8.23	10.15	12.15
80	5.37	6.01	6.87	8.00	9.94	11.85
90	5.15	5.79	6.57	7.80	9.80	11.79
100	4.93	5.51	6.36	7.51	9.52	11.50
105	4.79	5.46	6.22	7.37	9.37	11.37
110	4.65	5.28	6.15	7.22	9.23	11.15
115	4.50	5.15	5.93	7.08	9.08	11.07
120	4.36	4.93	5.79	6.93	8.95	10.94
125	4.15	4.79	5.64	6.79	8.80	10.80
130	3.86	4.65	5.44	6.57	8.65	10.58

TABLE 94.

DEGREES OF DIFFERENCE BETWEEN STEAM TEMPERATURE AND ACTUAL AVERAGE TEMPERATURE OF FEED WATER.

Vacuum Heaters between Engine and Condenser.

Initial Temperature of Water, t_0.	24″ Vacuum. Temperature 141° F. Final Temperature of Water.						25″ Vacuum. Temp. 134° F. Final Temperature of Water.			
	105	110	115	120	125	130	110	115	120	125
40	62.9	59.3	55.3	50.9	46.2	40.6	50.1	45.7	40.6	34.6
50	59.2	55.6	51.9	47.7	43.1	37.8	46.7	42.4	37.7	32.1
60	55.5	52.1	48.4	44.4	40.1	35.1	43.2	39.3	34.7	29.4
70	51.6	48.2	44.8	41.1	36.9	32.1	39.7	35.9	31.2	26.6
80	47.6	44.2	41.0	37.5	33.5	29.2	35.9	32.4	28.4	23.8

Initial Temperature of Water, t_0.	26″ Vac. Temp. 125° F. Final Temp. of Water.				27″ Vac. Temp. 114°. Final Temp. of Water.			28″ Vac. Temp. 100°. Final Temp. of Water.		
	105	110	115	120	90	100	105	70	80	90
40	42.0	40.4	35.0	28.3	44.4	36.0	30.8	43.2	36.3	27.9
50	39.9	37.3	32.3	25.9	40.8	32.8	28.0	39.1	32.7	24.8
60	38.2	34.1	29.4	23.4	36.9	29.6	25.1	34.7	28.8	21.6
70	34.3	30.8	26.9	20.7	33.0	26.2	22.1	......	24.7	18.1
80	30.8	27.3	23.3	18.1	28.7	22.5	18.7	......	......	14.4

Atmospheric Heaters.

Atmospheric Pressure. Temp. 212° F. Final Temperature of Water.						Initial Temperature of Water, t_0.
192	196	200	204	208	210	
70.6	65.7	60.1	53.5	44.8	38.2	40
67.9	63.1	57.6	51.2	42.8	36.4	50
65.1	60.4	55.2	48.9	40.7	34.7	60
62.2	57.7	52.6	46.6	38.7	32.9	70
59.4	54.9	50.0	44.2	36.6	31.0	80
56.4	52.3	47.4	41.8	34.4	29.1	90
53.1	49.3	44.7	39.3	32.4	27.3	100
51.9	47.9	43.4	38.2	31.4	26.4	105
50.3	46.4	42.1	36.9	30.2	25.5	110
48.8	45.0	40.6	35.7	29.2	24.5	115
47.2	43.5	39.2	34.4	28.0	23.5	120
45.6	41.9	37.8	33.1	26.9	22.5	125
43.9	40.3	36.4	31.7	25.8	21.5	130

velocity of the water and the material and character of the tubes. Increasing the velocity of the water passing through the heater increases the rate of heat transmission and thereby renders the heating surface more effective. In order to employ moderately high velocities and at the same time allow sufficient time in which to raise the temperature to a maximum, the tubes should be as long as practicable and of small diameters. Other things being equal, a heater containing a large number of tubes of small diameter is more economical than one containing a small number of large tubes. It is important to proportion the heater according to the amount of water to be heated and the maximum temperature to which the water must be raised. In designing a heater, then, the maximum amount of heat to be transmitted per degree difference in temperature per hour per square foot should be assumed, and the velocity of the water made such that it is capable of absorbing this amount. A good average figure for multi-flow heaters is U = 250 B.t.u. for plain brass or copper tubes and U = 300 B.t.u. for corrugated tubes with a water velocity of 50 feet per minute; for single-flow heaters, U = 175 (for plain brass) with a water velocity of 12.5 feet per minute and for coil heaters U = 300 (copper) with a water velocity of 150 feet per minute. These figures are for water-tube heaters only. For steam-tube heaters (iron tubes) a good average figure is U = 120. (See also Fig. 356.)

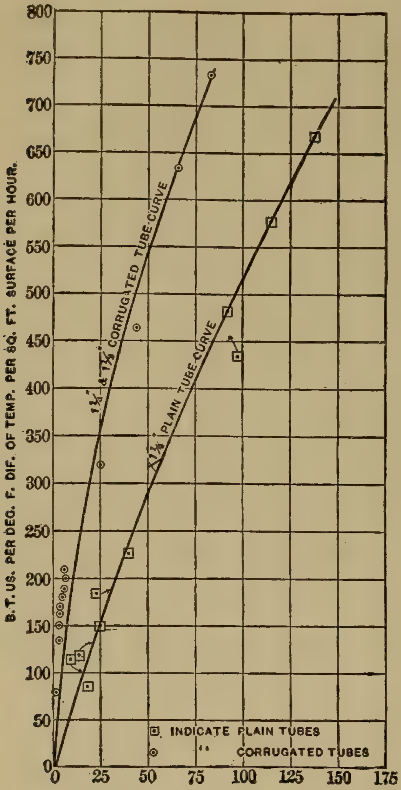


FIG. 356.

Experiments show that heaters and condensers operating with counter currents are more efficient and are capable of obtaining a higher final temperature than those operating with parallel currents. For a mathematical discussion of the parallel and counter-current flow see "Evaporating, Condensing and Cooling Apparatus," by E. Hausbrand, Chapter I.

It is generally supposed that the pressure of the steam and water in a closed heater has little influence on the heat transmission other than

that due to increased temperature difference, but experiments recently conducted seem to show that the pressure has a marked influence. Tests made by O. M. Row (Industrial Engineering, April, 1912, p. 314) seem to show that the transmission of heat from steam to water falls off as the pressure is increased, and that when the water and steam pressures become equal a further increase of water pressure has no effect. Similar results were obtained from experiments conducted by A. H. Tuells (Engineering, Feb. 23, 1912). The following results were obtained from the tests made by Row.

GALLONS (IMPERIAL) OF WATER PER HOUR RAISED 100° F. IN TEMPERATURE UNDER DIFFERENT STEAM AND WATER PRESSURES.

Water Pressure, Lbs. per Sq. In.	Steam Pressure, Lbs. Per Sq. In.		
	100	50	25
5	215	125	46
10	180	104	40
20	143	78	36
30	120	64	35
40	104	56	35
50	93	54	35
60	87	54	35
80	83	54	
100	82	54	

Transmission of Heat from Steam Through Surfaces: Engng., Feb. 9, 1912, p. 191.

Example: Determine the size of vacuum and atmospheric heaters for a condensing plant of 1200 i.h.p. Engines use 20 pounds of steam per i.h.p. hour; auxiliaries use the equivalent of 10 per cent of the main engine steam; vacuum 25 inches referred to 30-inch barometer; feed water, $t_0 = 50$ degrees; temperature of hot well, $t_2 = 110$ degrees; coefficient of heat transmission, $U = 300$ B.t.u.

Vacuum or Primary Heater.

Feed water for main engines,

$$20 \times 1200 = 24{,}000 \text{ pounds per hour.}$$

Feed water used by auxiliaries,

$$10 \text{ per cent of } 24{,}000 = 2400 \text{ pounds per hour.}$$

Total feed,

$$W = 24{,}000 + 2400 = 26{,}400 \text{ pounds per hour.}$$

From formula (207),

$$A = \frac{W}{U} \log_e \frac{t_s - t_0}{t_s - t_2}$$
$$= \frac{26{,}400}{300} \log_e \frac{134 - 50}{134 - 110}$$
$$= 110 \text{ square feet.}$$

On the basis of $\frac{1}{3}$ square foot of surface per horse power the rating of this heater will be

$$110 \times 3 = 330 \text{ horse power.}$$

Atmospheric or Secondary Heater.

The temperature of the feed water leaving the atmospheric heater, formula (197), will be

$$t = \frac{t_0 + 0.9\,S\,(H + 32)}{1 + 0.9\,S},$$

where $S = 0.10, \quad t_0 = 110 \text{ degrees}, \quad H = 1150 \text{ B.t.u.},$

whence
$$t = \frac{110 + 0.9 \times 0.10\,(1150 + 32)}{1 + 0.9 \times 0.10}$$
$$= 198 \text{ degrees.}$$

The required surface is

$$A = \frac{W}{U} \log_e \frac{t_s - t_0}{t_s - t_2},$$

where $t_s = 212, \quad t_0 = 110, \quad t_2 = 198,$

whence
$$A = \frac{26{,}400}{300} \log_e \frac{212 - 110}{212 - 198}$$
$$= 175 \text{ square feet.}$$

The horse-power rating will be

$$175 \times 3 = 525 \text{ horse power.}$$

295. Open vs. Closed Heaters. — Open and closed heaters have their respective advantages and a careful study of the various influencing conditions is necessary for an intelligent choice. The following parallel comparison brings out a few of the distinguishing features:

Open Heater.	Closed Heater.
Efficiency.	
With sufficient exhaust steam for heating, the feed water may reach the same temperature as the steam.	The maximum temperature of the feed water will always be 2 degrees or more lower than the temperature of the steam.
Scale and oil do not affect the heat transmission.	Scale and oil deposit on the tubes and the heat transmission is lowered.

Open Heater.	Closed Heater.
Pressures.	
It is not ordinarily subjected to much more than atmospheric pressure.	The water pressure is slightly greater than that in the boiler when placed on the pressure side of the pump as is customary.
Safety.	
Sticking of the back pressure valve may cause it to "blow up" if provision is not made for such an emergency.	It will safely withstand any pressure likely to occur.
Purification.	
Since the exhaust steam and feed water mingle, provision must be made for removing the oil from the steam. Scale and other impurities precipitated in the heater are readily removed.	Oil does not come in contact with the feed water. Scale is removed with difficulty.
Location.	
Must always be placed above the pump suction and on the suction side.	May be placed anywhere on the pressure side of the pump.
Pumps.	
With supply under suction two pumps are necessary and one must handle hot water.	One cold-water pump is necessary.
Adaptability.	
Particularly adaptable for heating systems where it is desired to pipe the "returns" direct to heater.	All vacuum or primary heaters are necessarily of this type.

296. "Through" Heaters. — Fig. 357 shows a typical installation of a through heater in a non-condensing plant.

It is evident that *all* the steam must pass *through* the heater. Now, one pound of exhaust steam in condensing gives up approximately 1000 B.t.u. Hence, if the initial temperature of the feed water is 50 degrees and the final temperature 210, the engine furnishes $\frac{1000}{210 - 50} = 6.26$, say, six times the quantity necessary for heating the feed water to a maximum. Therefore the area of the pipe supplying the heater with steam need be but one sixth that of the main exhaust. With the heater connected as in Fig. 357 the connections must necessarily be the same size as the exhaust pipe.

With this arrangement the heater cannot be "cut out" while the engine is in operation and hence it is not adapted for plants working

continuously. For the purpose of cutting out a heater while the plant is in operation a through heater may be by-passed as in Fig. 358. Advantage may be taken here of the permissible reduction in the size of pipes and fittings, i.e., valves, etc., at C and D need be but one half

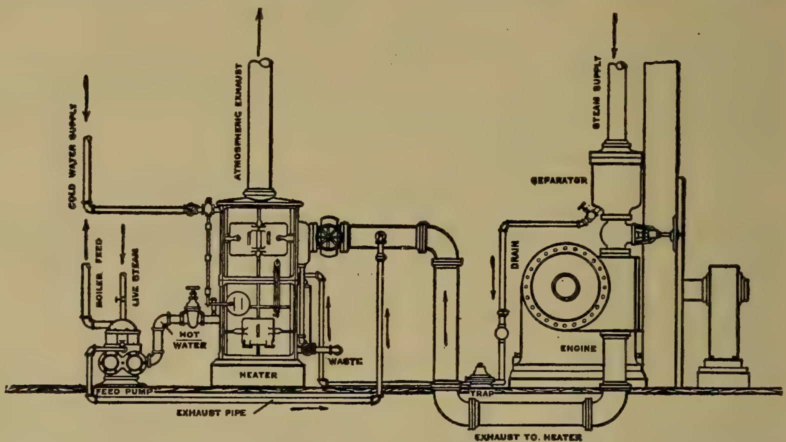


FIG. 357. Open Heater connected as a "Through" Heater. Non-condensing Plant.

the size of those at A. This reduction in size may prove to be a considerable item in large installations.

297. Induced Heaters. — Fig. 359 shows a typical installation of an induced heater in a non-condensing plant and Fig. 360 an induced primary heater in a condensing plant.

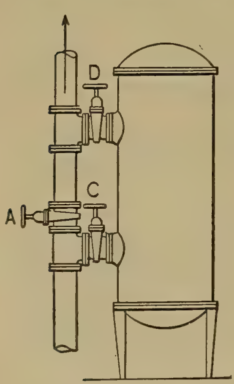


FIG. 358.

In the arrangement in Fig. 359 the number of fittings is reduced to a minimum and the heater may be readily cut out. Since induced heaters are apt to become air-bound, a vapor pipe or vent is inserted in the top of the heater as shown. This pipe varies from $\frac{1}{2}$ to $1\frac{1}{2}$ inches in diameter, depending upon the size of heater.

Closed Heaters: Am. Elecn., May, 1900, p. 236, July, 1900, p. 354, Oct., 1905, p. 530; Cassier's Mag., Aug., 1903, p. 330; Eng. U. S., Jan. 1, 1906, p. 13; Power, April, 1902, p. 11.

298. Live-steam Heaters and Purifiers. — The function of a live-steam heater and purifier is primarily that of purification and hence it is not ordinarily installed unless the feed water contains scale-forming elements such as sulphates of lime and magnesia. These, as previously stated, are not entirely precipitated until a temperature of approximately

300 degrees F. is reached; hence no amount of heating with exhaust steam at atmospheric pressure will thoroughly purify feed water containing these elements.

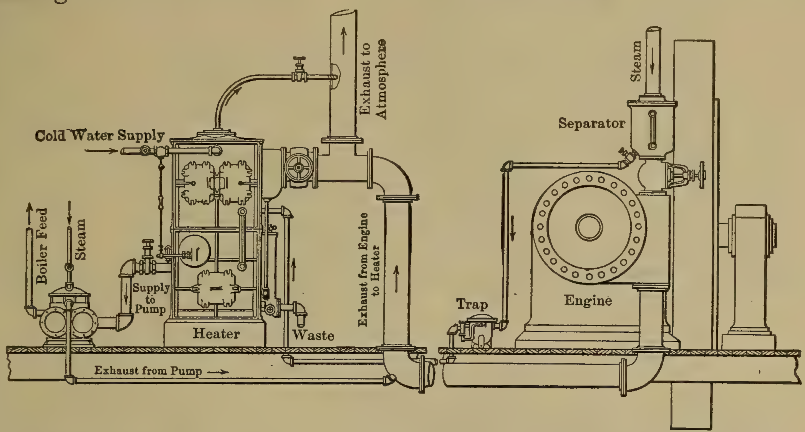


FIG. 359. Open Heater Connected as an "Induced" Heater. Non-condensing Plant.

Fig. 363 shows a section through a Hoppes live-steam purifier. Since the purifier is subjected to full boiler pressure, the shell and heads are constructed of steel. Within the shell are a number of trough-shaped

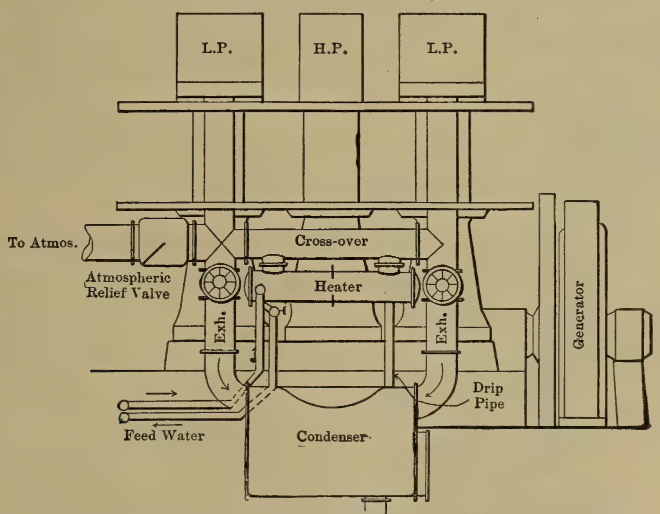


FIG. 360. Closed Heater Connected as an "Induced" Heater. Condensing Plant.

pans or trays placed one above another and supported on steel angle ways. Steam from the boiler enters the chamber at A and comes in contact with feed water and condenses. The water on entering the

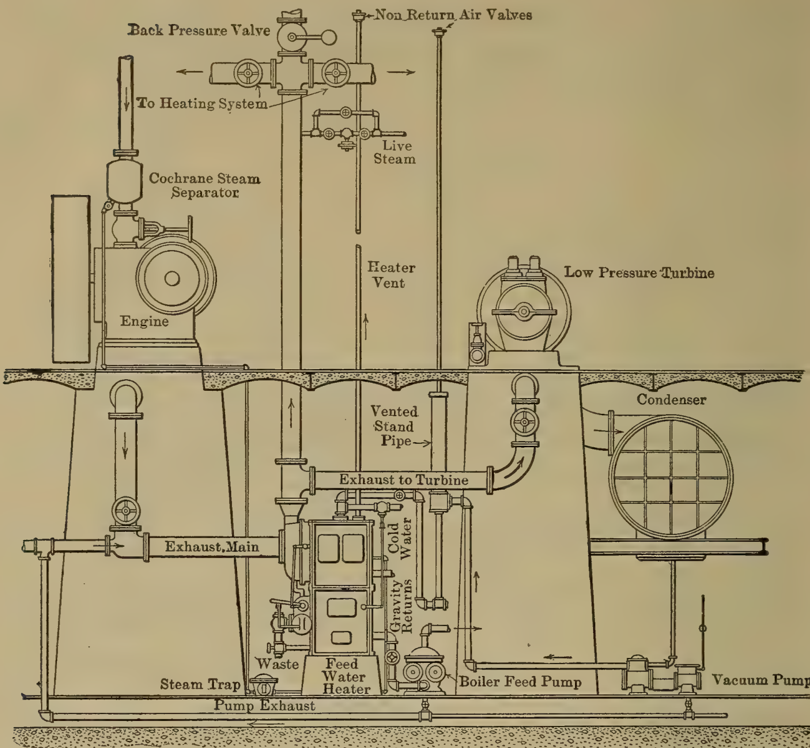


FIG. 361. Open Heater in Connection with a Low-pressure Turbine.

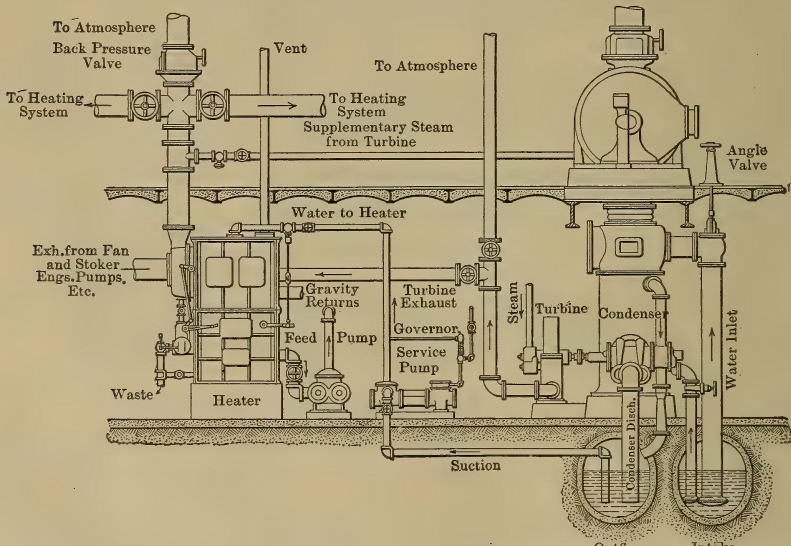


FIG. 362. Open Heater in Connection with a Jet Condenser.

heater at B is fed into the top pan and, overflowing the edges, follows the under side of the pan to the center and drops into the pan below. It flows over each successive pan in the same manner until it reaches

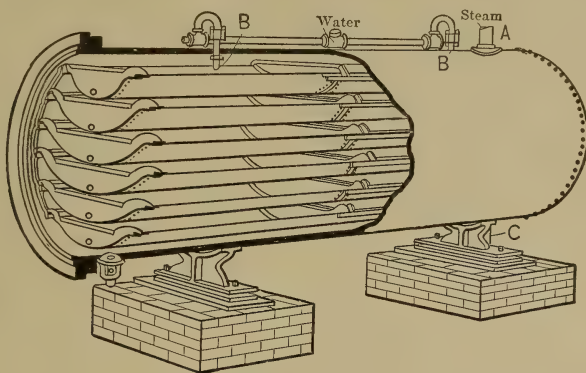


FIG. 363. Hoppes Live-steam Purifier.

the chamber at the bottom, whence it gravitates to the boiler through pipe C. As the steam inclosed in the shell comes in contact with the thin film of water, the solids held in solution are separated and adhere

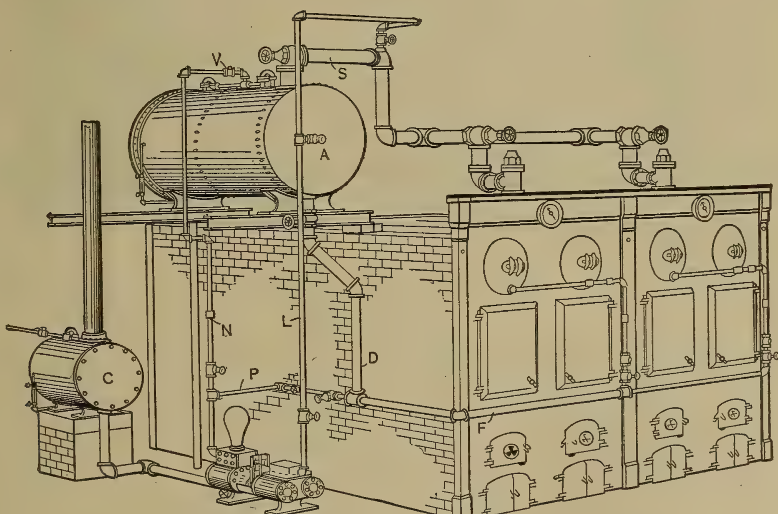


FIG. 364. Typical Installation of a "Live-steam" Purifier.

to the bottom of the pans in the same manner that stalactites form on the roofs of natural caves. Authentic tests show that live-steam heaters may increase the boiler efficiency. (See Power, Feb. 21, 1911, p. 295.) The purifier should be set in such a position as will bring the bottom of

the shell two feet or more above the water level of the boilers, as in Fig. 364. N is the feed pipe from pump to purifier and should be provided with a check valve. D is the gravity pipe through which the purified water flows to the boiler. This pipe should be carried below the water level of the boilers and all branch pipes should be taken off below the water line. Pipe L leads from top of pipe S to pump or other steam-using device. This is necessary in order that air and other non-condensable gases liberated from the water may be removed from the purifier, which would otherwise become air-bound. In the illustration the feed pump takes its supply from an exhaust steam heater C. The purifier is provided with a suitable by-pass so that the water may be fed directly to the boiler when necessary.

Live Steam Heated Feed Water: Elec. Engr., Lond., June 29, 1906; Cassier's Mag., Oct., 1911, p. 543; Elec. Rev., Lond., May 20, 1898, p. 667; Eng. Rec., Aug. 30, 1898, p. 467; Power, March 31, 1908, p. 498, Feb. 21, 1911, p. 295.

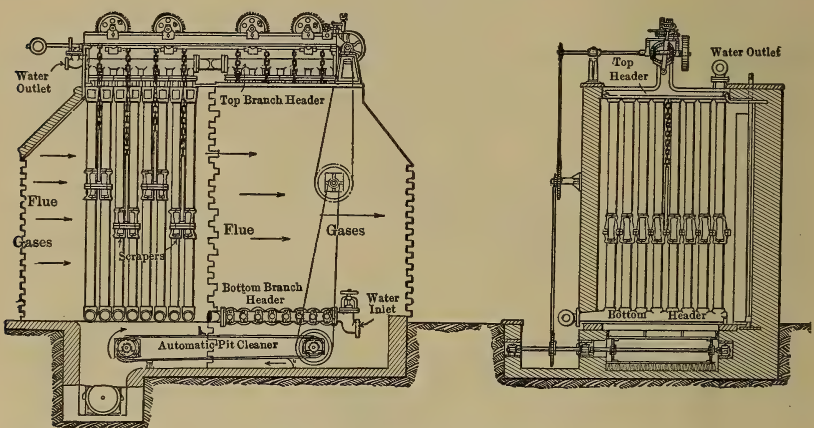


FIG. 365. Green Economizer.

299. Economizers. — Fig. 365 gives a general view of a Green economizer, illustrating a typical flue gas heater. It consists of a series of cast-iron tubes 9 to 10 feet in length and $4\frac{5}{8}$ inches in diameter, which are arranged vertically in sections of various widths across the main flue between boiler and chimney. When in position the sections are connected by top and bottom headers, and the headers are connected to branch pipes running lengthwise, one at the top and the other at the bottom. Both of the branch pipes are outside the brickwork which incloses the apparatus. The waste gases are led to the economizer by the ordinary flue from the boiler to the chimney, but a by-pass must be provided for use when the economizer is out of service for cleaning or for repairs. The feed water is forced into the economizer through the

lower branch pipe nearest the point of exit of gases, and emerges through the upper branch pipe nearest the point where the gases enter. Each tube is encircled with a set of triple overlapping scrapers which travel continuously up and down the tubes at a slow rate of speed, the object being to keep the external surfaces free from soot. The mechanism for working the scrapers is placed on top of the economizer, outside the chamber, and the motive power is supplied either by a belt from some convenient shaft or small independent engine or motor. The power for operating the gearing varies from 1 to $\frac{1}{2}$ horse power per 1000 square feet of economizer surface, depending upon the number and length of tubes. The apparatus is fitted with blow-off and safety valves, and a space is provided at the bottom of the chamber for the

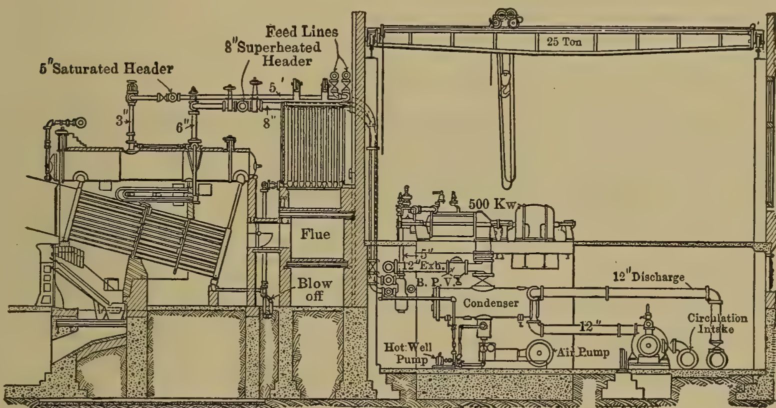


FIG. 366. Typical Economizer Installation.

collection of soot. For continuous plant operation the soot is automatically cleaned as shown in the illustration.

This type of economizer is also used as an *air heater* for drying and heating purposes. The air heater is similar in design to the water heater with the exception of the direction of flow and size of tubes. The tubes in the air economizer are $3\frac{7}{8}$ inches internal diameter by 9 feet in length, as against $4\frac{5}{8}$ inches internal diameter for the water economizer. In the latter the water enters at the bottom header and passes out from the top header, in the former the air is forced by a fan first through one set of tubes and up through another set, and then down again, and so on until it leaves the heater.

300. Value of Economizers. — The general conclusion drawn from current practice is that an economizer installation results in:

A small annual saving in cost of operating the plant.

Decreased wear and tear on the boilers due to the higher feed-water temperature.

A large storage of hot water for sudden increase in load.

Purification of the feed water due to the high temperature in the economizer. The scale-forming elements do not bake hard on the economizer tubes as they would in the boiler where the heat from the fire is more intense, but make a muddy deposit readily removed by blowing off.

301. Factors Determining Installation of Economizers. — The factors to be considered before installing an economizer are:

The nature of the auxiliary machinery, direct connected or belted.

Method of heating the feed water; whether vacuum and atmospheric heaters are used and whether all or part of the auxiliary steam is used for heating.

Initial temperature of the feed water; whether the feed is taken from the hot well or from a cold supply.

Rise in temperature due to economizer.

Cost of economizer. An approximate price is $15 per tube erected, on a basis of 15 square feet per tube. The heating surface is rated at 3 to 5 square feet per boiler horse power.

Cost of additional building space.

Reduction in boiler-heating surface made possible by the economizer.

Extra cost of stack or forced-draft apparatus necessary to compensate for loss of draft due to economizer. The economizer lessens the draft by increasing the resistance between boilers and chimney and by reducing the chimney temperature. Where the installation of an economizer decreases the normal temperature of the chimney from, say, 550 degrees to 350 degrees F., the reduction in draft is approximately 25 per cent.

Total cost of economizer plant. This depends largely upon the design and varies from $4 to $7 per boiler horse power.

Interest, depreciation, repairs, operation, taxes, and insurance.

Table 95 gives the results of economizer tests.

302. Temperature due to Use of Economizer. — The rise in temperature of feed water due to the use of an economizer may be approximated from the following empirical formula advocated by the Green Economizer Company:

$$x = \frac{y\,(T_1 - t_1)}{9.1 + \dfrac{5\,w + GC}{2\,GC}\,y}, \tag{213}$$

in which

x = rise in temperature of the feed water,
T_1 = temperature of flue gas entering economizer,

t_1 = temperature of feed water entering economizer,
w = pounds of feed water per boiler horse power per hour,
G = pounds of flue gas per pound of combustible,
C = pounds of coal per boiler horse power per hour,
y = square feet of economizer heating surface per boiler horse power.

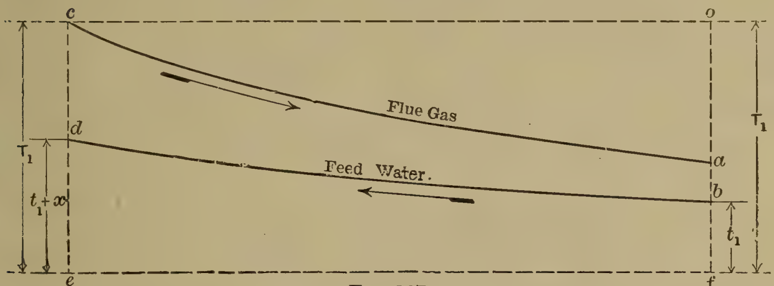


FIG. 367.

Referring to Fig. 367, the ordinates represent temperatures, and abscissas the path of the flue gas and the water in the economizer. The flue gas enters the economizer at c with temperature T_1 and leaves at a with temperature T. The feed water enters at b with temperature t_1 and leaves at d with temperature $t_1 + x$.

The algebraic mean temperature difference D between the flue gas and the feed water will be

$$D = \frac{cd + ab}{2} \tag{214}$$

$$= \frac{T_1 - (t_1 + x) + of - oa - bf}{2} \tag{215}$$

$$= \frac{T_1 - (t_1 + x) + T_1 - oa - t_1}{2} \tag{216}$$

$$= T_1 - t_1 - \frac{x + oa}{2}.^* \tag{217}$$

Now, wx = B.t.u. absorbed by the feed water per boiler horse power and

GCS = B.t.u. given up to the feed water by the flue gas for each degree reduction in temperature (S = mean specific heat of the flue gas); therefore

* The theoretical mean temperature difference is $d = \dfrac{t_1 - t_2}{\log_e \frac{t_1}{t_2}}$ in which t_1 = original temperature difference and t_2 = final temperature difference between the two fluids.

$wx \div GCS$ = total reduction in temperature of the flue gas, that is,

$$\frac{wx}{GCS} = oa. \tag{218}$$

Substituting (218) in (217), we get

$$D = T_1 - t_1 - \frac{x + \dfrac{wx}{GCS}}{2} \tag{219}$$

$$= T_1 - t_1 - x\frac{GCS + w}{2\,GCS}, \tag{220}$$

in which

D = mean temperature difference between flue gas and feed water, degrees F.

Let U = B.t.u. absorbed per hour per square foot per degree difference in temperature and

y = square feet of economizer surface per boiler horse power.

Then $U\,Dy$ = heat absorbed per boiler horse-power hour.

But wx = heat absorbed per boiler horse-power hour.

Therefore $$wx = U\,Dy. \tag{221}$$

Combining (220) and (221),

$$wx = Uy\left(T_1 - t_1 - \frac{x\,GCS + wx}{2\,GCS}\right), \tag{222}$$

from which $$x = \frac{y\,(T_1 - t_1)}{\dfrac{w}{U} + \dfrac{w + GCS}{2\,GCS}}. \tag{223}$$

y varies from 3.5 square feet to 5 square feet per boiler horse power, and U from 2.25 to 3.3, depending upon the conditions of operation.*

If we let $w = 30$, $S = 0.2$, and $U = 3.3$, and substitute these values in equation (223), it assumes the form given by the Green Economizer Company, equation (213).

A method of approximating the rise in temperature where the final temperature of the flue gas is known, is to assume $\frac{5}{10}$ degree rise in the feed water for each degree reduction in temperature in the flue gas. This is determined on the basis that approximately 20 pounds of flue gas are generated for each pound of combustible, and that 10 pounds of water are evaporated per pound of combustible; that is, 2 pounds of flue gas are generated for each pound of feed water delivered to the boiler. Assuming a specific heat of 0.25 for the flue gas, this gives 2×0.25, or 0.5 degree rise in temperature in the feed water for each degree reduction in the flue gas temperature.

* For $D = 600$	$U = 3.25$	For $D = 400$	$U = 2.75$
500	3.00	300	2.25

Example: Determine the rise in temperature of the feed water in a power plant of 1200 i.h.p. Engines use 20 pounds of steam per i.h.p. hour; auxiliaries use the equivalent of 12 per cent of main engine steam; vacuum 25 inches; feed-water supply 50 degrees; 3.7 pounds of coal are burned per hour per boiler horse power; flue gas temperature 550 degrees F.; steam pressure 150 pounds gauge.

The vacuum and atmospheric heater will raise the temperature of the feed water from 50 to 205 degrees. (See preceding problem.)

On the assumption that 20 pounds of flue gas are generated per pound of combustible and that 3.5 square feet of economizer heating surface are installed per boiler horse power, the notations in the formula will become $T_1 = 550$, $t_1 = 205$, $w = 30$, $G = 20$, $C = 3.7$, $y = 3.5$, $U = 3.3$, $S = 0.2$.

Substituting,

$$x = \frac{3.5\,(550 - 205)}{\frac{30}{3.3} + \frac{(5 \times 30 + 20 \times 3.7)}{(2 \times 20 \times 3.7)}3.5}$$

= 83 degrees rise in temperature.

Therefore the temperature of the water entering the boiler will be

$$205 + 83 = 288 \text{ degrees F.}$$

Economizers: Prac. Engr. U. S., March, 1910, May, 1910, p. 282, July 15, 1912, p. 736; Power, July 27, 1909; Cassier's, March, 1900, p. 378; Eng. Mag., June, 1912, p. 389.

TABLE 95.

ECONOMIZER PERFORMANCES.

Number of Plant.	Number of Economizer Tubes Installed.	Temperatures, Deg. Fah.					
		Gases Entering Economizer	Gases Leaving Economizer	Fluid Entering Economizer	Fluid Leaving Economizer	Rise in Temperature of Fluid.	Actual Saving in Fuel, Per Cent.
Water Heater.							
1	160	435	279	84.2	196.2	112.0	12.5
2	....	416	254	40.0	185.4	125.4	13.8
3	960	620	293	101.0	237.0	136.0	18.3
4	520	548	295	96.0	200.0	104.0	9.2
5	520	603	325	93.5	203.8	110.3	9.7
6	384	368	245	103.0	202.6	99.6	12.4
7	448	537	326	71.2	203.4	132.2	17.5
Air Heater.							
1	72	301	257	70.0	152.0	82.0	....
2	240	512	319	54.0	201.6	147.6	9.0
3	96	557	376	41.0	200.0	159.0	14.0
4	192	417	369	74.0	210.0	136.0	....

Compiled from "The Book of the Economizer," 1912, published by the Green Engineering Co.

303. Choice of Feed-water Heating System. — The heating of feed water and its delivery to the boiler in the most economical manner is a problem involving such a large number of combinations that a general analysis is impracticable. The following discussion of a specific case will give some idea of the manner in which this problem may be attacked.

Example: Determine the most economical manner of heating the feed water for a power plant of 1000 horse power operating under the following conditions: Schedule 10 hours per day and 310 days per year; load factor on the ten-hour basis 0.8; cost of coal $2.50 per ton of 2000 pounds; heat value of the coal 13,500 B.t.u. per pound; average boiler efficiency 65 per cent; engines use 20 pounds of steam per i.h.p. hour; steam pressure 150 pounds absolute; temperature of cold water 60 degrees; vacuum 26 inches referred to 30-inch barometer; interest 5 per cent; depreciation $8\frac{1}{2}$ per cent; maintenance 1 per cent; insurance $\frac{1}{2}$ per cent; taxes 1 per cent; total charges 16 per cent; charges for attendance and maintenance assumed to be the same in each case and credit for the chimney assumed to offset debit for economizer space. Many of the influencing conditions are left out for the sake of simplicity.

The most likely combinations are

(1) Atmospheric, all auxiliaries steam driven, water taken from cold well.

(2) Same as (1) except that water is taken from hot well.

(3) Economizers, auxiliaries electrically driven, chimney draft, water from cold well.

(4) Vacuum heater, economizer, and electrically driven auxiliaries, fan draft.

(5) Vacuum heater, atmospheric heater, and steam auxiliaries.

(6) Atmospheric heater, economizer, steam auxiliaries, fan draft.

(7) Vacuum and atmospheric heaters, economizers, steam auxiliaries, and electrical fan.

(8) Vacuum, atmospheric heater, economizer, and chimney draft, auxiliaries operating condensing except feed pumps and stoker engines which exhaust into the atmospheric heater.

The difference between the total heat furnished by the boiler and the heat returned in the feed water is the net heat put into the steam by the boiler. Evidently the system which shows the least net heat required to produce one horse power will be the most economical as

far as coal consumption is concerned, although not necessarily the cheapest when both operating and fixed charges are considered.

Prices vary so much that it is practically impossible to give costs of installations which will bear criticism and the prices taken in this problem are approximate only.

Case I.

Atmospheric heater, auxiliaries steam driven, feed from cold well.

This arrangement and that of Case II are the most common in power plants of this size.

The power consumption of the auxiliaries operating non-condensing varies from 8 to 12 per cent of the total power developed. Assume it to be 10 per cent.

The temperature of the feed water leaving the heater may be determined by formula (197).

$$t = \frac{t_0 + 0.9\,S\,(\lambda + 32)}{1 + 0.9\,S}.$$

Substituting $S = 0.10,\quad \lambda = 1146,\quad t_0 = 60,$

$$t = \frac{60 + 0.9 \times 0.10\,(1146 + 32)}{1 + 0.9 \times 0.10}$$

$$= 152.$$

The *net heat* furnished by the boiler to produce one indicated horsepower hour in the engine is evidently the heat necessary to raise 20 + 10 per cent of 20 = 22 pounds of water from 152 degrees F. to steam at 150 pounds pressure; i.e., the net heat furnished is

$$22 \times 1071.2 = 23{,}564 \text{ B.t.u.}$$

Now, 1 i.h.p. = 2546 B.t.u.

Therefore the heat efficiency of this arrangement is

$$\frac{2546}{23{,}564} = 10.8 \text{ per cent.}$$

Probable First Cost.

Steam pumps	$ 400.00
Condenser with steam-driven air and circulating pumps	3000.00
1000-horse-power open heater	480.00
Piping	1200.00
	$5080.00

Fuel Consumption.

Average horse-power hours per year = 1000 (rated horse power) $\times$ 0.8 (curve load factor) $\times$ 310 (days per year) $\times$ 10 (hours per day) = 2,480,000.

Pounds of coal per i.h.p. hour = net heat furnished per i.h.p. hour $\div$ net heat absorbed by the boiler per pound of coal = 23,564 $\div$ (13,500 $\times$ 0.65) = 2.68.

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.68}{2000} = 3323.$$

Fuel and Fixed Charges.

Fuel, 3323 tons at $2.50	$8308.00
Fixed charges, 16 per cent of $5080	812.00
	$9120.00

CASE II.

Same as Case I, except that feed is taken from the hot well. This arrangement is possible only when the condensing water is suitable for feed purposes.

Assume the temperature of the water from the hot well as it enters the heater to be 110 degrees.

The temperature of the feed water leaving the heater will then be 198 degrees (from formula (197)).

$$\text{Net heat furnished} = 22 \times 1025.2 = 22{,}554 \text{ B.t.u.}$$

$$\text{Efficiency} = \frac{2546}{22{,}554} = 11.3 \text{ per cent.}$$

$$\text{Pounds of coal per i.h.p. hr.} = \frac{22{,}554}{13{,}500 \times 0.65} = 2.62.$$

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.62}{2000} = 3248.$$

Fuel and Fixed Charges.

Fuel, 3248 tons at $2.50	$8120.00
Fixed charges (same as Case I)	812.00
	$8932.00

CASE III.

Economizers, auxiliaries electrically driven, chimney draft, water from the cold well.

Practice gives an average of 3 per cent of the main engine output as the power required to operate the electrical auxiliaries in a plant of this size.

The temperature of the feed water leaving the economizer may be determined from formula (213).

$$x = \frac{y\,(T_1 - t_1)}{9.1 + \frac{5\,w + GC}{2\,GC}\,y}.$$

Substituting,

$$x = \frac{3.5\,(550 - 60)}{9.1 + \frac{5 \times 30 + 20 \times 3.7}{2 \times 20 \times 3.7}\,3.5} = 119 \text{ degrees.}$$

Temperature of feed water entering heater $= 119 + 60 = 179$ degrees.

Net heat furnished $= (20 + 3$ per cent of $20) \times 1044.2 = 21{,}510$ B.t.u.

$$\text{Efficiency} = \frac{2545}{21{,}510} = 11.8 \text{ per cent.}$$

Probable First Cost.

Economizers	\$3500.00
Motor feed pump	600.00
Condenser with electrically driven air and circulating pump	6000.00
Piping and wiring	1000.00
	\$11,100.00

Fuel Consumption.

$$\text{Pounds of coal per i.h.p. hour} = \frac{21{,}510}{13{,}500 \times 0.65} = 2.45.$$

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.45}{2000} = 3038.$$

Fuel and Fixed Charges.

Fuel, 3038 tons at \$2.50	\$7595.00
Fixed charges, 16 per cent on \$11,100	1776.00
	\$9371.00

Case IV.

Vacuum heater, economizer, electrically driven auxiliaries, fan draft.

The vacuum heater may be relied upon to raise the temperature of the feed water to 110 degrees.

The economizer will increase this 107 degrees (from formula (213)), giving the feed water a temperature of 217 degrees as it enters the heater.

The electrical fan for the mechanical-draft system will require approximately 2 per cent of the main system engine power, making a total of $3 + 2 = 5$ per cent for all auxiliaries.

$$\text{Net heat furnished} = (20 + 5 \text{ per cent of } 20) \times 1006.2$$
$$= 21{,}130 \text{ B.t.u.}$$

$$\text{Efficiency} = \frac{2545}{21{,}130} = 12.05 \text{ per cent.}$$

Probable First Cost.

For the sake of simplicity it is assumed that the high first cost of the chimney plus its low depreciation and maintenance will offset the low first cost of the mechanical-draft system plus its higher maintenance and depreciation charges:

Economizers	$3500.00
Motor feed pump	600.00
Motor-driven pumps and condenser	6000.00
Motor-driven fan	750.00
Piping and wiring	1200.00
Vacuum heater	200.00
	$12,250.00

Fuel Consumption.

$$\text{Pounds of coal per i.p.h. hour} = \frac{21{,}130}{13{,}500 \times 0.65} = 2.41.$$

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.41}{2000} = 2988.$$

Fuel and Fixed Charges.

Fuel, 2988 tons at $2.50	$7470.00
Fixed charges, 16 per cent of $12,250	1960.00
	$9430.00

In like manner Cases V, VI, VII, and VIII have been treated and are tabulated in the summaries.

SUMMARY (1).

Case.	Temperature of Feed Water.	Power Consumed by Auxiliaries.	Efficiency.	First Cost.	Fuel Cost per Year.	Cost of Operation per Year.
	Degrees F.	Per Cent.	Per Cent.			
I	152	10	10.8	$5,080	$8,308	$9,120
II	198	10	11.3	5,080	8,120	8,932
III	179	3	11.8	11,100	7,595	9,371
IV	217	5	12.05	12,250	7,470	9,430
V	208	10	11.4	5,280	7,900	8,744
VI	294	14	12	9,000	7,750	9,190
VII	290	10	12.2	9,300	7,380	9,570
VIII	270	8	12.3	8,250	7,075	8,395

SUMMARY (2).

Case.	Efficiency.	First Cost.	Fuel.	Cost per Year.
I	8	1	8	4
II	7	1	7	2
III	6	6	4	6
IV	3	7	3	7
V	5	2	6	3
VI	4	4	5	5
VII	2	5	2	8
VIII	1	3	1	1

Summary (2) gives the ranking; thus: Case I is eighth in point of efficiency; first in cheapness of installation; eighth in yearly cost of fuel; and fourth in yearly cost of operation. Case VIII is apparently the best arrangement for the *given conditions.*

CHAPTER XIII.

PUMPS.

304. Classification. — Pumps used in connection with steam power plants may be conveniently classified under five groups according to the principles of action.

1. *Piston pumps,* in which motion and pressure are imparted to the fluid by a reciprocating piston, plunger, or bucket. The action is positive and a certain definite amount of fluid is handled per stroke under predetermined conditions of pressure and velocity.

2. *Centrifugal pumps,* in which the fluid is given initial velocity and pressure by a rotating impeller. The action is not positive, as the amount of fluid discharged is not necessarily proportional to the impeller displacement.

3. *Rotary pumps,* in which motion and pressure are imparted to the fluid by a rotating impeller. The volume discharged is practically equal to the impeller displacement regardless of pressure.

4. *Jet pumps,* in which velocity and pressure are imparted to the fluid by the momentum of a jet of similar or other fluid. The ordinary steam injector is the best known of this group.

5. *Direct-pressure pumps,* in which the pressure of one fluid acts directly on the surface of another fluid, thereby imparting all or part of its energy to the latter. The pulsometer is an example of this type.

These groups may be variously subdivided as follows:

Piston	Direct-acting	Simplex, Duplex	Air. Vacuum. Forcing. Lifting.
	Fly-wheel, Power driven	Simplex, Duplex, Triplex	
Centrifugal	Volute	Single stage	Vacuum. Forcing. Lifting.
	Turbine	Multi-stage	
Rotary	Power driven	Forcing, Lifting	
Jet	Injector	Positive, Automatic	
Direct pressure	Pulsometer	Lifting	
	Air-lift	Lifting	

Piston or *plunger pumps* are the most common in use. Boiler-feed pumps, city waterworks pumps, and force pumps are ordinarily of this type. In the direct-acting type, Fig. 369, the water plunger and steam piston are secured to a single piston rod and the steam pressure

is transmitted directly to the water. There is no flywheel, connecting rod, or crank. The velocity of the delivery is proportional to the resistance offered by the water; when the resistance equals the forward effort of the steam pressure the pump stops. This class of pump is well adapted for boiler-feeding purposes, since it may be operated as slowly as suits the requirements of feeding by simply throttling the discharge. The steam consumption is very large in proportion to the work performed, since the steam is not used expansively.

Flywheel pumps, Figs. 382, 423, are ordinarily classified as pumping engines. In this class steam may be used expansively, as sufficient energy is stored in a flywheel to permit the drop in steam pressure during expansion. These pumps find wide application in city waterworks, elevator plants, and the like, where high duty is required. They are little used as stationary boiler feeders, but are used to some extent in river-boat practice and in plants operating continuously for long periods at comparatively steady loads. Practically all sizes of dry-air pumps and a number of large jet condenser pumps are of this type.

Piston pumps, Fig. 389, driven by gearing or belting are ordinarily classified as power-driven pumps. The driving power may be steam engine, electric motor, or gas engine. The single-cylinder machine is often designated as a "simplex" power-driven pump, the two-cylinder as a "duplex," the three-cylinder as a "triplex," and so on.

Centrifugal pumps, Fig. 409, are supplanting to a considerable extent the present type of piston pump for many uses. Though particularly adapted for low heads and large volumes they are used in many situations requiring extremely high heads. They are not as efficient as high-grade pumping engines, but the extremely low first cost frequently offsets this disadvantage, and they are much used in connection with dry docks, irrigating plants, sewage systems, and as circulating and vacuum pumps in condensing plants.

Rotary pumps, Fig. 421, are employed to a limited extent in the same field as the centrifugal pump. Being positive in action, they permit of a much lower rotative speed for the same delivery pressure.

Jet pumps, Fig. 393, are seldom used as pumps in the ordinary sense of the word, on account of their extremely low efficiency, but are frequently employed for discharging water from sumps. Their greatest field of application lies in boiler feeding and in this respect their efficiency is comparable with that of the average piston pump.

Direct-pressure pumps operated by steam, such as the "pulsometer," Fig. 424, are used principally for pumping out sumps, surface drains, and the like, where the operation is intermittent. Direct-pressure pumps of the air-lift type, Fig. 425, are quite common and are used a

great deal in situations where water is to be pumped from a number of scattered wells.

305. Boiler-feed Pumps, Direct-acting Duplex. — Figs. 368 and 369 illustrate a typical duplex boiler-feed pump, which consists virtually of two direct-acting pumps mounted side by side, the water ends and the steam ends working in parallel between inlet and exhaust pipe. The piston rod of one pump operates the steam valve of the other through the medium of bell cranks and rocker arms. The pistons move alternately, and one or the other is always in motion, the flow of water being practically continuous.

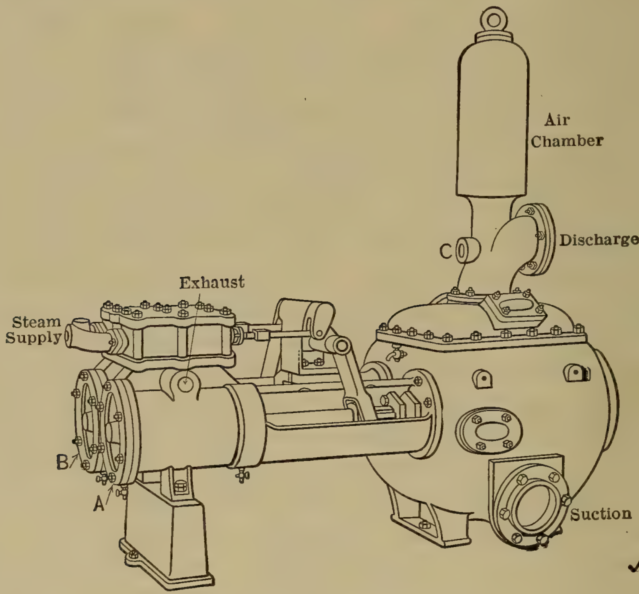


Fig. 368. Typical Duplex Pump.

In general construction the steam pistons and valves are similar to those of steam engines. The valves in duplex pumps, however, have no lap. In order to reduce the valve travel to a minimum, and still have sufficient bearing surface between the steam ports and the main exhaust ports to prevent the leakage of steam from one to the other, separate exhaust ports are provided which enter the cylinder at nearly the same point as the steam ports. This arrangement offers a simple means of cushioning the piston by exhaust steam, thus preventing it from striking the cylinder heads at the ends of the stroke. The valves of the duplex pump having no lap would, if connected rigidly to the valve stem, open one port as soon as the other had been closed, at about mid-stroke of the piston, thus cutting down the stroke

to about one fourth the usual length. To obviate this difficulty the valves are given considerable lost motion by allowing sufficient clearance between the lock nuts on the valve stem; the latter, therefore, imparts no motion to the valve until the piston operating it has nearly

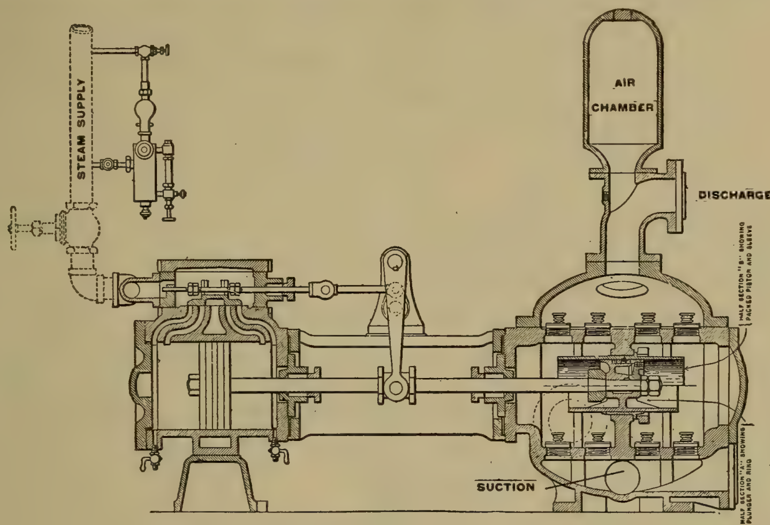


FIG. 369. Section through a Typical Duplex Boiler-feed Pump.

completed the stroke. The lost motion between valves and lock nuts renders it impossible to stop the pump in any position from which it cannot be started by simply admitting steam, and therefore the pump has no dead centers. When one piston moves to the end of the stroke it pulls or pushes the opposite valve to the end of its travel; then when the piston starts back to the other end of its stroke the valve remains stationary, owing to the lost motion, until the piston has completed about one half the stroke. During this time the opposite piston has completed a full stroke and the valve operated by it will have opened the steam port wide, so that while one valve covers both steam ports the other is at the end of its travel. In some makes of pumps the stem is rigidly attached to the valves, the lost motion being adjusted outside the steam chest as shown in Figs. 370 and 371, which represent two common constructions of duplex valve gear.

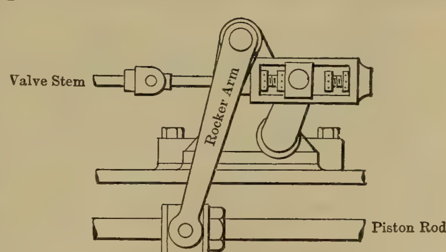


FIG. 370.

Fig. 372 shows the valve and piston in the position occupied at the commencement of the stroke. At one end of the valve the steam port *P* is open wide and at the opposite end the exhaust port *E* is open wide. When the piston nears the opposite end of the stroke and reaches the position shown in Fig. 373 the steam escape through the exhaust port *E* is cut off by the piston, and since the steam port is closed, the remaining steam is compressed between the piston and cylinder head, thus arresting the motion of the piston gradually without shock or jar.

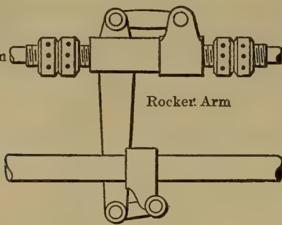


Fig. 371.

The construction of the water end of single-cylinder and duplex pumps is practically the same; any slight differences which may be found are confined to minor details which in no way affect the general design or operation of the pump. The piston is double acting, the single-acting cylinder being confined to power pumps or to steam pumps intended for very high pressures. In the old-style pumps it was the custom to use one large valve with a lift sufficient to give the required passage, but in modern practice the required area is divided among several small valves, so that each one is easily and cheaply removed in case of accident or wear, and slip is lessened.*

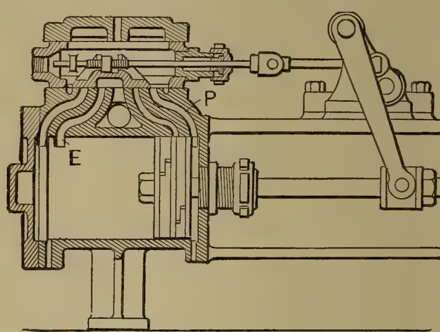


Fig. 372.

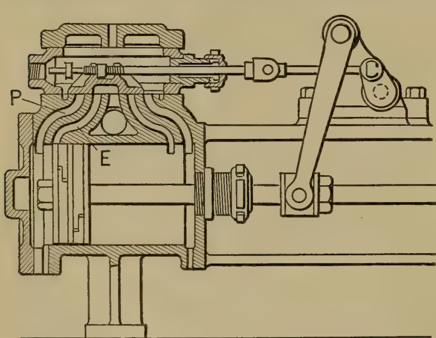


Fig. 373.

The valves are carried by two plates or decks, the suction valves being attached to the lower plate and the delivery valves to the upper one, as shown in Fig. 369.

The valves in practically all boiler-feed pumps are of the flat disk type, Fig. 374, held firmly to the seat by conical springs and guided by a bolt through the center.

* The modern Riedler pump is an exception. See Engineer, U. S., Nov. 15, 1907, p. 1040.

All pumps are provided with an air chamber on the discharge side, which acts as a cushion for the water, prevents excessive pounding, and insures a uniform flow. Fig. 375 shows a section through the steam end of a compound duplex pump.

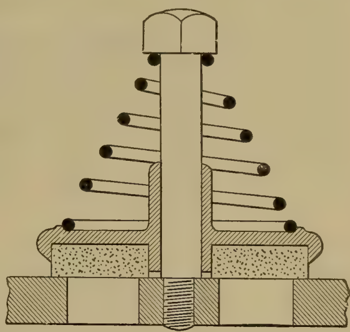


FIG. 374. A Typical Pump Disk Valve.

306. Feed Pumps with Steam-actuated Valves. — Single-cylinder direct-acting pumps, Fig. 376, are ordinarily operated by steam-actuated valves. The steam enters the chest *C* and passes to the left through the annular opening *A* formed between the reduced neck of the valve and the bore of the steam chest. It is thus projected against the inside surface of the valve head *H* before escaping through the port *P* and passing to the cylinder. Both the pressure and impulse due to velocity acting on the valve head *H* tend to close or restrict the admission port by forcing the valve to the left. On reaching the cylinder and forcing the piston *X* toward the right, the pressure of the steam upon

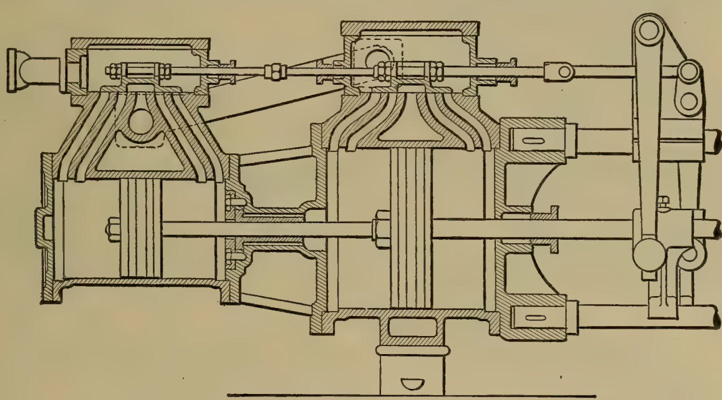


FIG. 375. Section through Steam Cylinders of a Typical Compound Duplex Pump.

the opposite side of the valve head *H* is pressing the valve to the right, a movement which would give the admission more port opening at *A* and deliver more steam to the cylinder. The valve then holds a position depending upon the relative intensity of the two pressures, which tend to move it in opposite directions, the admission steam, tending to close the valve, and cylinder steam, tending to open the valve wider. The steam valve, therefore, is always in a balanced position. The steam piston is grooved at the center, forming a reservoir for live steam *R* which is supplied from the upper chamber of the steam chest by passage

E to the cylinder cap *S*, and thence by tube *M* and the hollow piston rod *V*. The steam in this annular piston space reverses the steam valve by pressing alternately against the outer surfaces of the valve heads *H* through the connecting passages *O*, *O* near each end of the cylinder. The tappets *T* are for the purpose of moving the valve by hand in case it fails to move automatically. Steam-actuated valves are not as positive in action as mechanically operated valves, and hence are little used in situations where positive action is essential, as in fire-pump service.

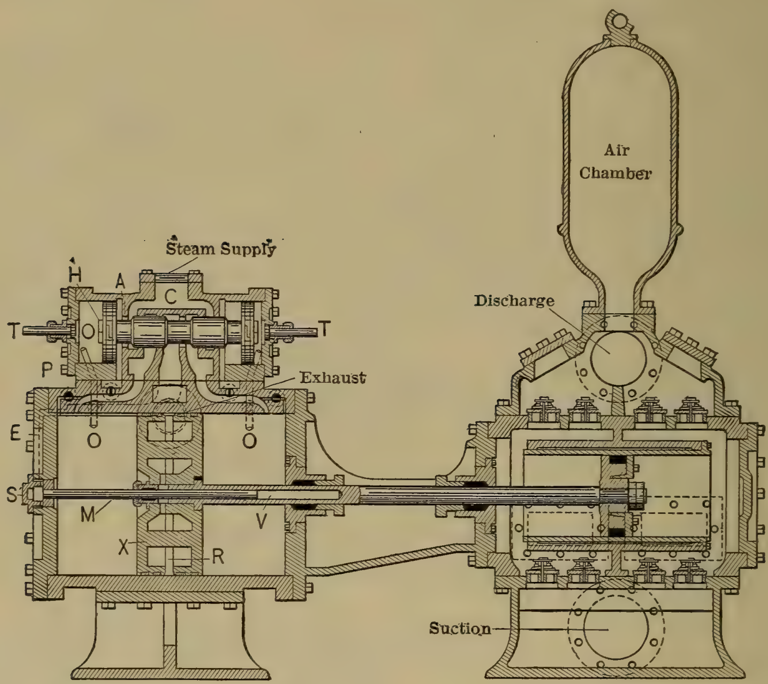


FIG. 376. Marsh Boiler-feed Pump. A Typical Steam-actuated Valve Gear.

307. Air and Vacuum Chambers. — Air chambers in piston pumps are for the purpose of causing a steady discharge of water and of reducing excessive pounding at high speeds by providing a cushion for the water. The water discharged under pressure compresses the air in the air chamber somewhat above the normal pressure of discharge during each stroke of the water piston, and when the piston stops momentarily at the end of the stroke the air expands to a certain extent and tends to produce a uniform rate of flow.

The volume of the air chamber varies from 2 to $3\frac{1}{2}$ times the volume of the water piston displacement in single-cylinder pumps, and from

1 to $2\frac{1}{2}$ times in the duplex type. High-speed pumps are provided with air chambers of from 5 to 6 times the piston displacement. The water level in the air chamber should be kept down to one fourth the height of the chamber. In slow-running pumps sufficient air may be carried into the pump chamber along with the water, but with high speeds a large part of the air will be discharged, and air must be forced into the chamber by mechanical means. The larger the chamber the more uniform will be the discharge pressure.

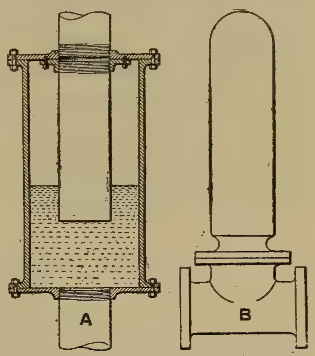


FIG. 377. Forms of Vacuum Chambers.

Vacuum chambers are frequently provided for the purpose of maintaining a uniform flow of water in the suction pipe and assisting in the reduction of slip. Such chambers should be of slightly greater volume than the suction pipe and of considerable length rather than diameter. Fig. 377 illustrates two designs commonly used. The one in Fig. 377 (B) should be placed in such a position as to receive the impact of the

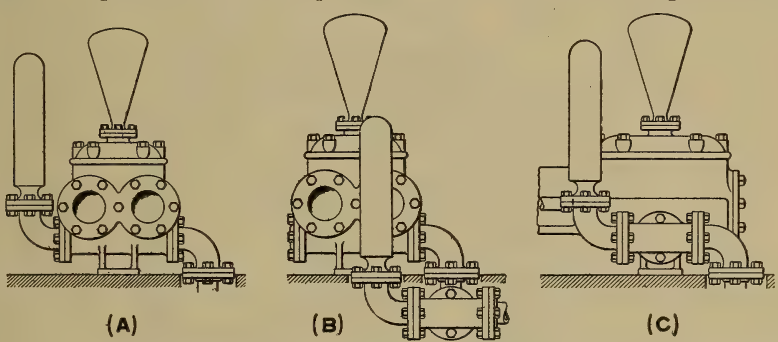


FIG. 378. Different Arrangements of Vacuum Chambers.

column of water in the suction pipe as illustrated in Fig. 378 (A), (B) and (C). The chamber illustrated in Fig. 377 (A) should be placed in the suction pipe below but close to the pump.

308. Water Pistons and Plungers. — In cold-water pumps the water pistons are usually packed with some kind of soft packing. Fig. 379 (A) shows the details of a piston with square *hydraulic packing*. The body E is fastened to the piston rod by nut C; packing is placed at D, and follower F is forced up by the nut B and locked by nut A. For large sizes the design is the same except that the follower is set up by a number of nuts near the edge. In hot-water pumps the pistons are often packed by means of *metallic piston rings* R, R, Fig. 379 (C), similar

to those in steam pistons, or merely by *water grooves G, G,* Fig. 379 (B). The water end is often fitted with a *plunger* instead of a piston, as in Figs. 380 to 382. The piston is more compact, but the plungers do not require a bored cylinder, so that the first cost is not materially different.

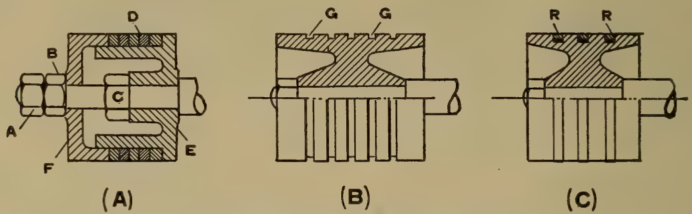


FIG. 379. Types of Water Pistons.

Fig. 380 shows a plunger with metal packing ring. When leakage becomes excessive it is necessary to renew the ring, which is readliy removed.

In Fig. 381 the plunger is packed with hydraulic packing as in the follower type of pump piston. The great difficulty with the above

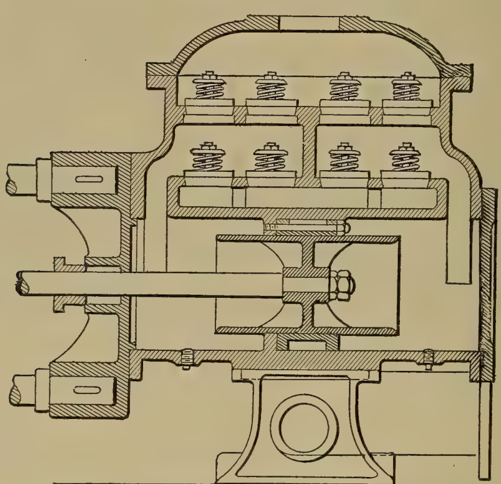


FIG. 380. Plunger with Metal Packing Ring.

types of piston and plunger is in keeping the packing tight or in knowing when it is leaking, and the trouble necessary to replace the packing. The *outside packed plunger,* Fig. 382, obviates these disadvantages to a great extent, since leakage is readily detected and repacking is performed without removing the cylinder heads. In dirty, dusty locations, however, the piston pump or inside packed plunger is to be preferred, since the abrasive action of the dust renders outside packing difficult.

Fig. 382 illustrates a high-duty elevator pump with outside packed plunger.

309. Performance of Piston Pumps. — Direct-acting pumps as a class are wasteful of fuel and low in efficiency, due largely to the non-expansive use of steam. The average small duplex boiler-feed pump uses from 100 to 200 pounds of steam per i.h.p. hour, depending upon the speed, and the mechanical efficiency varies from 50 per cent to 90 per cent. When new and in proper working condition the mechanical efficiency is seldom less than 85 per cent; but such pumps, as a rule, are given scant attention, and the average efficiency is not far from

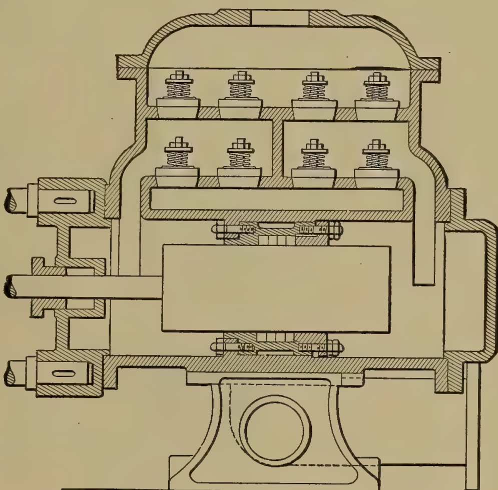


Fig. 381. Plunger with Hydraulic Packing.

65 per cent. The term "mechanical efficiency" in this connection refers to the ratio of the actual water horse power to the indicated horse power of the steam cylinder. The loss includes the slip of the piston and valves. A steam consumption of 150 pounds per i.h.p. hour with mechanical efficiency of 65 per cent is equivalent to a power consumption of about 5 per cent of the rated boiler capacity, although if the exhaust steam is used for feed-water heating the actual heat consumption may be but 1 to 1.5 per cent. Compound direct-acting pumps running non-condensing use from 50 to 100 pounds of steam per i.h.p. hour. Single-cylinder flywheel pumps of the slow-speed type, running non-condensing, use about 50 pounds of steam per i.h.p. hour. Multi-cylinder flywheel pumps of the high-duty type use about 25 pounds per i.h.p. hour when running non-condensing, and as low as 10 pounds when operating condensing. High-grade *direct-connected* motor-driven

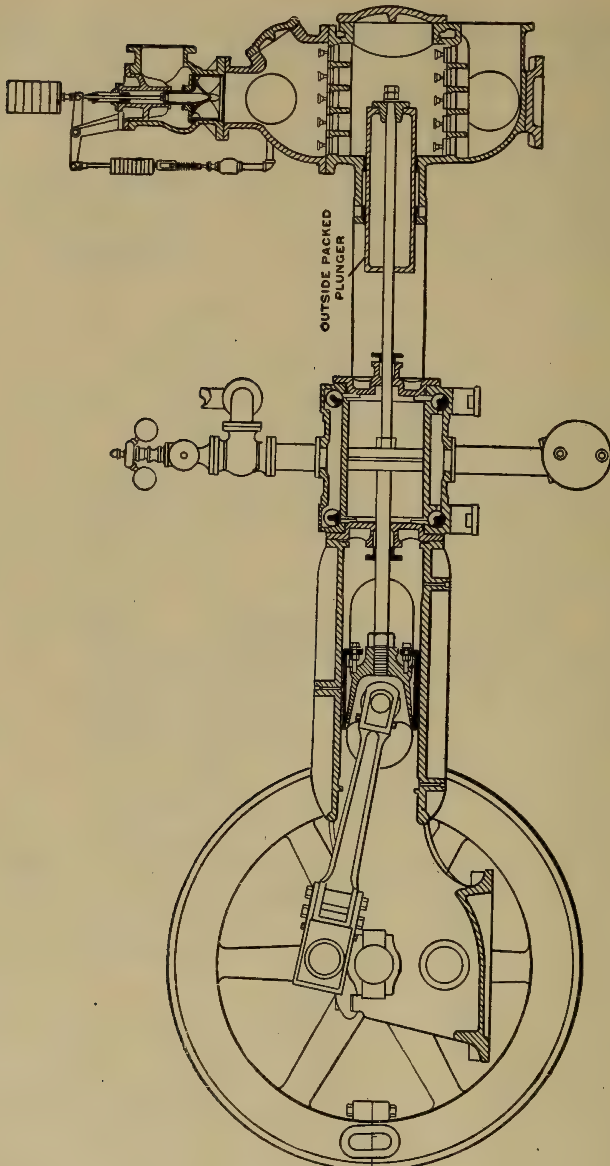


FIG. 382. Horizontal **Fly-Wheel Pump** with Outside Packed Plunger.

power pumps have a mechanical efficiency from line to water load, at normal rating, of about 80 per cent. The efficiency of *geared* pumps at normal rating varies with the character of the gearing and the degree of speed reduction, and may range anywhere from 40 to 70 per cent.

The steam consumption of all direct-acting boiler pumps decreases with the increase in speed. This is illustrated by curve *B*, Fig. 383, plotted from the tests of a 12 × $7\frac{1}{4}$ × 12 direct-acting single-cylinder pump at Armour Institute of Technology, and curve *A* based on experiments with a 16 × 12 duplex fire pump at Massachusetts Institute of Technology.

Fig. 384 gives the details of the performance of a 12 × $7\frac{1}{4}$ × 12 Marsh boiler-feed pump at the Armour Institute of Technology.

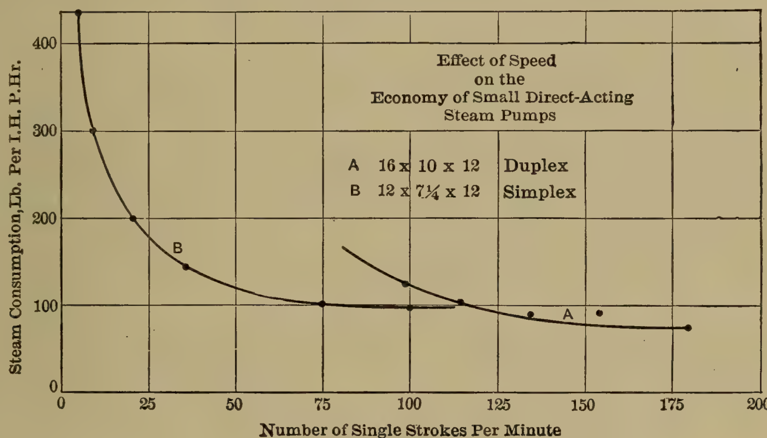


FIG. 383.

The determination of the power consumption of a boiler-feed pump is best illustrated by the following example.

Example: A small direct-acting duplex pump uses 150 pounds of steam per i.h.p. hour. Gauge pressure 150 pounds per square inch; feed-water temperature 64 degrees F. Required the per cent of rated boiler capacity necessary to operate the pump.

The head pumped against, 150 pounds per square inch, is equivalent to 150 × 2.3 = 345 feet of water.

The friction through the valves, fittings, and pipe, and the vertical distance between suction and feed-water inlet, are assumed to be equivalent to 20 per cent of the boiler pressure, giving a total head of 150 + 30 = 180 pounds per square inch, or 414 feet of water.

A boiler horse power, taking into consideration leakage losses and the steam used by the feed pump, will be equivalent to the evaporation

of approximately 32 pounds of water per hour from a feed temperature of 64 degrees F. to steam at 150 pounds gauge.

The actual work done in pumping 32 pounds of water against a head of 414 feet is

$$414 \times 32 = 13{,}248 \text{ foot-pounds.}$$

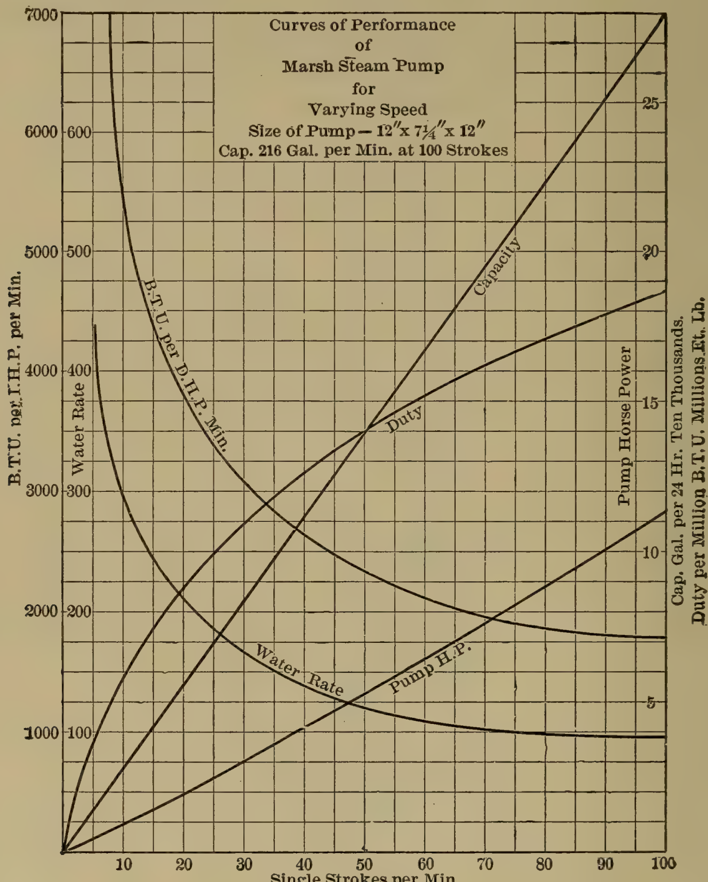


FIG. 384.

This corresponds to

$$\frac{13{,}248}{60 \times 33{,}000} = 0.0067 \text{ horse power.}$$

The total heat of one pound of steam above 64 degrees F. is 1163 B.t.u. The heat delivered to the pump per i.h.p. hour is

$$1163 \times 150 = 174{,}450 \text{ B.t.u.}$$

The amount used by the pump for each boiler horse power, disregarding efficiency, is

$$174{,}450 \times 0.0067 = 1168 \text{ B.t.u. per hour.}$$

The mechanical efficiency of the average feed pump ranges from 50 to 85 per cent, depending upon its condition and the number of strokes per minute. Assuming it to be 65 per cent, the heat used by the pump per hour to deliver 32 pounds of water into the boiler is

$$1168 \div 0.65 = 1796 \text{ B.t.u.}$$

A boiler horse power is equivalent to 33,479 B.t.u. per hour. Therefore the per cent of boiler output necessary to operate the pump is

$$100 \times \frac{1796}{33{,}479} = 5.36 \text{ per cent.}$$

If the exhaust steam is used for heating the feed water, the steam consumption will be 0.73 per cent of the boiler capacity, thus: The weight of steam consumed per boiler-horse-power hour

$$\frac{1796}{1163} = 1.54 \text{ pounds.}$$

Allowing a 10 per cent loss, the heat in the exhaust available for heating the feed water is

$$[1150 - (64 - 32)]\, 0.9 \times 1.54 = 1550 \text{ B.t.u.}$$

$1796 - 1550 = 246$ B.t.u., or the net heat required by the pump per hour to deliver 32 pounds of water to the boiler.

The per cent of boiler output necessary to operate the pump is

$$100 \frac{246}{33{,}479} = 0.73.$$

Pump performances are generally given in terms of the foot-pounds of work done by the water piston per thousand pounds of dry steam or per million B.t.u. consumed by the engine, thus:

1. $$\text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Weight of dry steam used}} \times 1000. \qquad (234)$$

2. $$\text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Total number of heat units consumed}} \times 1{,}000{,}000. \qquad (235)$$

TABLE 96.

PERFORMANCE OF STEAM PUMPS. DUTY IN MILLIONS OF FOOT-POUNDS PER MILLION B.T.U.

$$\text{Mechanical efficiency} = \frac{\text{Pounds discharged per min.} \times \text{head in feet}}{\text{I.H.P. of steam cylinder} \times 33{,}000}.$$

Steam Consumption, Lbs. per I.H.P. Hour.		0.95	0.90	0.85	0.80	0.75	0.70	0.65	0.60	0.55	0.50
Initial Press. 100 Lbs. Gauge, Non-Cond.	200	9.40	8.91	8.42	7.92	7.42	6.93	6.44	5.94	5.45	4.95
	190	9.90	9.39	8.86	8.36	7.83	7.31	6.79	6.27	5.74	5.22
	180	10.45	9.90	9.38	8.80	8.25	7.70	7.15	6.60	6.05	5.50
	170	10.90	10.50	9.90	9.32	8.74	8.15	7.57	6.99	6.41	5.83
	160	11.75	11.13	10.51	9.90	9.28	8.66	8.04	7.42	6.81	6.19
	150	12.55	11.90	11.22	10.59	9.90	9.29	8.49	7.92	7.26	6.60
	140	13.45	12.75	12.02	11.32	10.61	9.90	9.20	8.49	7.76	7.07
	130	14.49	13.71	12.96	12.20	11.42	10.69	9.90	9.15	8.38	7.62
	120	15.67	14.85	14.03	13.20	12.38	11.55	10.71	9.90	9.08	8.25
	110	17.10	16.21	15.31	14.40	13.50	12.60	11.70	10.80	9.90	9.00
	100	18.81	17.82	16.82	15.82	14.84	13.86	12.88	11.88	10.89	9.90
	90	20.90	19.80	17.76	16.72	15.67	14.63	13.58	12.54	11.49	10.45
	80	23.51	22.27	21.03	19.80	18.56	17.32	16.08	14.85	13.61	12.37
	70	26.90	24.50	24.04	22.64	21.22	19.80	18.40	16.98	15.58	14.14
	60	31.35	29.70	28.05	26.40	24.75	23.10	21.43	19.80	18.15	16.50
	50	37.62	35.64	33.64	31.68	29.68	27.72	25.76	23.76	21.98	19.80
	40	47.02	44.55	42.07	39.60	37.12	34.65	32.17	29.70	27.22	24.75
	30	62.70	59.40	56.10	52.80	49.50	46.20	42.86	39.60	36.30	33.00
Initial Press. 150 Lbs. G. Vacuum 2 Lbs. Abs.	25	68.40	64.80	61.20	57.60	54.00	50.04	46.80	43.20	39.60	36.00
	20	85.50	81.00	76.50	72.00	67.50	63.00	58.50	54.00	49.50	45.00
	18	95.00	90.00	85.00	80.00	75.00	70.00	65.00	60.00	55.00	50.00
	16	106.87	101.25	95.62	90.00	84.37	78.75	73.12	67.50	61.87	56.25
	15	114.00	108.00	102.00	96.00	90.00	84.00	78.00	72.00	66.00	60.00
	14	122.14	115.71	109.29	102.86	96.43	90.00	83.57	77.14	70.71	64.28
	13	131.53	124.61	117.61	110.77	103.84	96.92	90.00	83.07	76.15	69.22
	12	142.50	135.00	127.50	120.00	112.50	105.00	97.50	90.00	82.50	75.00
	11	155.45	147.30	139.00	130.90	122.70	114.50	106.30	98.16	90.00	81.80
	10	171.00	162.00	153.00	144.00	135.00	126.00	117.00	108.00	99.00	90.00

(See A.S.M.E. code for conducting duty trials of pumping engines, Trans. A.S.M.E., 12–530, 563.) See also, Appendix F.

Example: A compound feed pump uses 100 pounds of steam per i.h.p. hour; indicated horse power, 48; capacity, 400 gallons per minute; temperature of water, 200 degrees F.; total head pumped against, 175 pounds per square inch; steam pressure, 100 pounds gauge; moisture in the steam, 3 per cent. Required the duty on the dry steam and on the heat-unit basis.

175 pounds per square inch is equivalent to $175 \times 2.4 = 420$ feet of water at 200 degrees F.

Weight of 400 gallons of water at 200 degrees F. $= 400 \times 8.03 = 3212$ pounds.

Work done per minute $= 3212 \times 420 = 1{,}349{,}040$ foot-pounds.

Weight of dry steam supplied per minute

$$= \frac{100 \times 48}{60} \times 0.97 = 77.6 \text{ pounds.}$$

B.t.u. supplied per minute

$$= \frac{100 \times 48}{60} (0.97 \times 879.8 + 309 - 200 + 32) = 79{,}552.$$

Duty per thousand pounds of dry steam

$$= \frac{1{,}349{,}040}{77.6} \times 1000 = 17{,}384{,}150 \text{ foot-pounds.}$$

Duty per million B.t.u.

$$= \frac{1{,}349{,}040}{79{,}552} \times 1{,}000{,}000 = 16{,}958{,}000 \text{ foot-pounds.}$$

Table 96 may be used in approximating the duty, thus:

The mechanical efficiency of the pump in the preceding problem is

$$\text{Efficiency} = \frac{\text{p.h.p.}}{\text{i.h.p.}} = \frac{1{,}349{,}040}{33{,}000 \times 48} = 85 \text{ per cent.}$$

At the intersection of vertical column "85" and horizontal column "100" of Table 96, we find 16.82 millions. See also, Table 65.

Tables 97 and 98 give the maximum theoretical height to which pumps may lift water by suction at different temperatures. In practice these figures cannot be realized. It is customary to have the water gravitate to the pump for all temperatures over 120 degrees F.

TABLE 97.

MAXIMUM HEIGHT TO WHICH PUMPS CAN RAISE WATER BY SUCTION.

(Temperature of Water 40 degrees F.; Barometer 29.92.)

Vacuum in Suction Pipe, Inches of Mercury.	Theoretical Lift.	Probable Actual Lift.	Vacuum in Suction Pipe, Inches of Mercury.	Theoretical Lift.	Probable Actual Lift.
	Feet.	Feet.		Feet.	Feet.
1	1.1	0.9	16	18.0	14.4
2	2.2	1.8	17	19.1	15.3
3	3.3	2.7	18	20.2	16.1
4	4.5	3.6	19	21.4	17.1
5	5.6	4.5	20	22.5	18.0
6	6.7	5.4	21	23.7	18.9
7	7.9	6.3	22	24.8	19.8
8	9.0	7.2	23	25.9	20.7
9	10.1	8.1	24	27.0	21.6
10	11.3	9.0	25	28.2	22.7
11	12.4	9.9	26	29.3	23.9
12	13.5	10.8	* 27	30.4	24.3
13	14.6	11.7	28	31.6	25.2
14	15.8	12.6	29	32.7	26.1
15	16.9	13.5	† 29.68	33.6	....

* Vacua greater than 27 inches are practically unobtainable in pumping practice except in connection with condensers.

† Maximum theoretical vacuum obtainable with water at 40 degrees F. and barometer of 29.92 inches.

TABLE 98.

MAXIMUM THEORETICAL HEIGHT TO WHICH A PUMP CAN LIFT WATER BY SUCTION AT DIFFERENT TEMPERATURES.

(Barometer 29.92.)

Temperature of Feed Water.	Maximum Theoretical Lift.	Temperature of Feed Water.	Maximum Theoretical Lift.
Degrees F.	Feet.	Degrees F.	Feet.
40	33.6	130	29.2
50	33.5	140	27.8
60	33.4	150	25.4
70	33.1	160	23.5
80	32.8	170	20.3
90	32.4	180	16.7
100	31.9	190	12.8
110	31.3	200	7.6
120	30.3	210	1.3

310. Size of Boiler-feed Pump.

Let D = diameter of water cylinder, inches.
d = diameter of the steam cylinder, inches.
L = length of stroke, inches.
N = number of working strokes per minute.
H = head in feet between suction and boiler water level.
R = resistance in pounds per square inch between suction level and boiler water level due to valves, pipes, and fittings.
p = boiler pressure, pounds per square inch.
S = ratio of the water actually delivered to the piston displacement.
W = weight of water delivered, pounds per hour.
I = indicated horse power of the pump at maximum capacity.
E = mechanical efficiency of the pump, taken as the ratio of the water horse power at the discharge opening to the indicated horse power of the pump, steam end.

Then

$$W = \frac{\pi}{4}\cdot\frac{D^2}{144}\cdot\frac{LN}{12} \times 60 \times 62.5 \times S = 1.7\, D^2LNS. \tag{236}$$

$$D = 0.77\sqrt{\frac{W}{LNS}}. \tag{237}$$

$$d = D\sqrt{\frac{p + R + 0.433\, H}{Ep}}. \tag{238}$$

$$I = \frac{W\,(p + R + 0.433\, H)\, 2.3}{33{,}000 \times 60 \times E}. \tag{239}$$

In average practice the piston or plunger displacement is made about twice the capacity found by calculation from the amount of water required for the engine, to allow for leakage, steam consumption of the auxiliaries, blowing off, and pump slip.

For pumps with strokes of 12 inches or over, the speed of the plunger or piston is usually limited to 100 feet per minute as a maximum to insure smooth running. For shorter strokes a lower limit should be used. The maximum number of strokes ranges from 100 for strokes over 12 inches in length, to 200 for strokes under 5 inches. Boiler-feed pumps should be designed to give the desired capacity at about one-half the maximum number of strokes or less.

Pump slip varies from 2 to 40 per cent, depending upon the condition of the piston and valves and the number of strokes. An average value for piston and plunger pumps in first-class condition is 8 per cent when operating at rated capacity, but it is wise to allow a much larger figure, say 20 per cent, for leakage caused by wear.

The area of the steam cylinder is made from 2 to 2.5 times that of the water end to allow for the various friction losses and the drop in pressure between the pump throttle and the boiler. The total head pumped against includes the suction lift, the friction of valves and fittings, the distance between the suction inlet and the boiler level and the boiler pressure. The excess head varies in practice from 15 to 40 per cent of the boiler pressure; an average figure is 25 per cent. In allowing for the drop in steam pressure between boiler and pump a liberal figure is 25 per cent.

The application of formulas (236) to (239), including the practical considerations stated above, is best illustrated by a specific example.

Example: Determine the size of direct-acting single-cylinder feed pump necessary to supply water to 1000 horse power of boilers. Gauge pressure 100 pounds per square inch; feed-water temperature 150 degrees F.

One horse power is equivalent to the evaporation of 34.5 pounds of water from and at 212 degrees F.; but the pump is usually designed to supply about twice the capacity.

Thus $W = 62{,}400$ (under the given conditions).
$S = 0.8$ (by assumption).
$LN = 1200$ (on the basis of 100 feet per minute).

Substitute these values in (237):

$$D = 0.77\sqrt{\frac{62{,}400}{1200 \times 0.8}} = 6.2 \text{ inches, — call it 6 inches,}$$

since the assumptions have been very liberal.

Assume $(0.433\,H + R) = 0.25\,p$ and $E = 0.65$.

Substitute these values in (238):

$$d = 6\sqrt{\frac{100 + 25}{0.65 \times 100}}$$
$$= 8.35, \text{— call it 8.5 inches.}$$

Allowing 100 strokes per minute the length of the stroke must be

$$L = 1200 \div 100 = 12 \text{ inches.}$$

The dimensions of the pump are $8\frac{1}{2} \times 6 \times 12$.

The indicated horse power at maximum load may be obtained by substituting the proper values in (239), thus:

$$I = \frac{62{,}400\,(100 + 25)\,2.3}{33{,}000 \times 60 \times 0.65}$$
$$= 13.9 \text{ i.h.p.}$$

311. Steam-pump Governors. — Fig. 385 shows a section through a Fisher pump governor, illustrating a device for maintaining a practically constant pressure in the discharge pipe irrespective of the quantity of water flowing. It embodies a pressure-reducing valve in the steam supply pipe of the pump, actuated by the slight variations in water pressure. When the demand for water increases, the pressure in the discharge pipe tends to decrease, and this drop in pressure (transmitted to the pump governor by suitable piping) causes more steam to be admitted, which increases the speed of the pump. The governor is connected to the steam inlet of the pump at *B* and the steam enters at *A*. Double-balanced valve *C* regulates the supply of steam to the cylinder by the amount it is raised from the seat. The valve is held open by spring *G*, the compression of which may be regulated by hand wheel *K*. The water pressure from the discharge pipe acts on piston *F* and tends to overcome the resistance of the spring. The difference in pressure between the water and the spring determines the position of valve *C*.

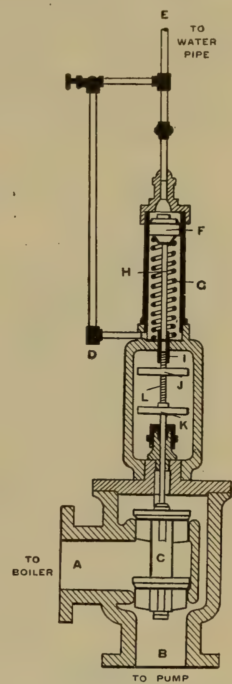


FIG. 385. Fisher Pump Governor.

Piston rod *H* is pinned to sleeve *I* and valve stem *L* screwed into this sleeve by means of hand wheel *K*. Hence, during ordinary operation, the piston, piston-rod sleeve, valve stem, and valve act as a single unit. By turning the hand wheel *K*, valve stem *L* will screw into sleeve *I* and the tension on the spring will be increased. Hand wheel *J* serves as a lock nut and prevents *K* from turning during normal operation.

312. Feed-water Regulators.— The water level in the boiler should be kept as nearly constant as possible, and this necessitates considerable attention on the part of the fireman, especially with fluctuating loads. There are a number of devices on the market which are designed to automatically maintain a constant level, and in many small plants where the duties of the fireman are numerous such devices in connection with high and low water alarms are of considerable assistance. Their action, however, is not always positive on account of wear or sticking of parts, and engineers as a rule prefer to rely upon hand regulation.

Fig. 386 shows a section through a Kitts feed-water regulator, consisting of two parts, the chamber *F* and the regulating valve *V*. The

float chamber is connected to the boiler or water column at *O* and *E*, and the regulating valve to the feed main at *R* and to the boiler feed pipe at *W*. When the water in the boiler falls below the mean level, the weight *B* overcomes the counterweight *G* and closes needle valve *L* by means of compound levers. At the same time an extension on valve *L* lifts spring *A* and opens exhaust valve *D*. This removes the steam pressure from the top of diaphragm *C*, in the regulating valve, through the agency of pipe *K*. The pressure from the pump raises the disk *T* and water flows into the boiler until the water rises to the mean level.

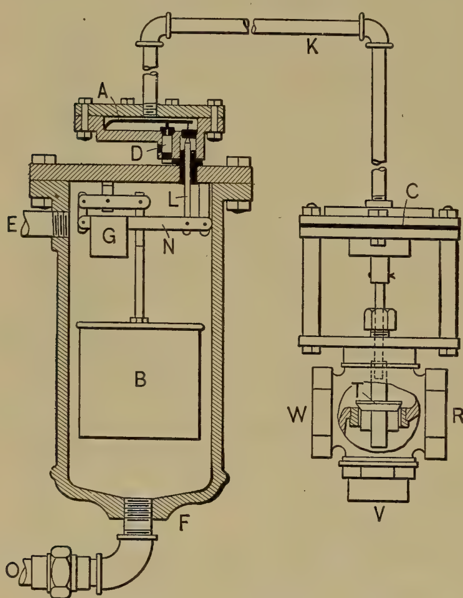


FIG. 386. Kitts Feed-water Regulator.

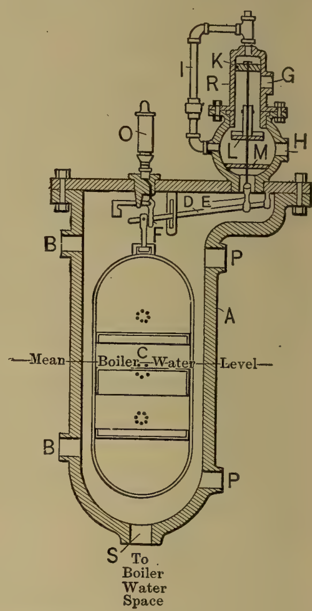


FIG. 387. Rowe Feed-water Regulator.

When weight *B* becomes submerged its weight is overcome by counterweight *G*, valve *L* is opened and exhaust valve *D* is closed. This admits steam pressure to the diaphragm *C* and forces disk *T* to its seat, cutting off the supply of water to the boiler.

The Rowe feed-water regulator, Fig. 387, depends for its operation on a familiar float-controlled valve mechanism. The vessel *A* is connected to the boiler above and below the water line, and the float *C*, following the water level up and down, actuates a balanced valve in accordance with the boiler-feed requirements. When this apparatus is used to regulate the feed of a single boiler the opening *G* in the valve chamber is connected to the steam space of the boiler and the outlet *H* is carried to the steam inlet of the feed-water pump. When the water

level is normal the float closes the valve L and thereby cuts off the supply of steam to the pump cylinders. Communication between chambers A and R is prevented by means of a diaphragm M. When the water level falls below normal the float pulls the valve down, opening the way for steam to pass from the inlet G to the outlet H and thence to the pump. When the regulator is used to control a battery of boilers the pump discharge delivers into the inlet G and the water passes through H to the boiler-feed main. Should the water level fall

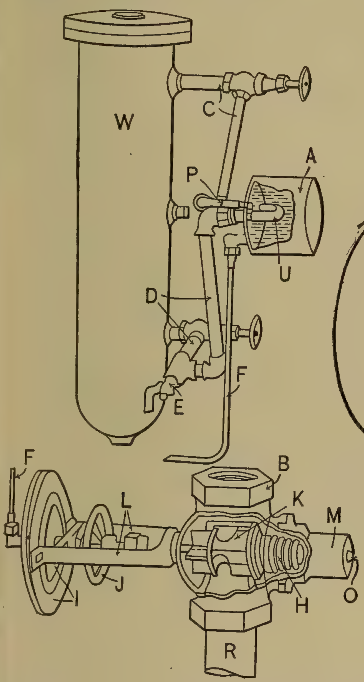


FIG. 388. "S–C" Feed Water Regulator.

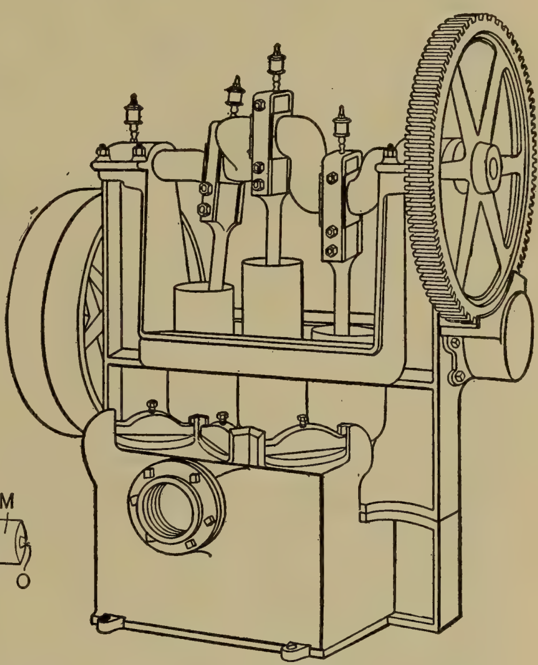


FIG. 389. A Typical Geared Triplex Pump.

beyond a predetermined limit by reason of any accidental discontinuance of the water supply which the apparatus cannot correct, the float would open the valve F of the alarm whistle O mounted on the top of the main vessel.

Fig. 388 gives the general details of the "S–C" feed water regulator which differs from the types just described in the manner of actuating the water regulating valve. A small copper vessel, A, partly filled with water, is in communication with diaphragm I through the medium of tube F. The water in vessel A is independent of the boiler supply. A small copper U-tube, U, projects into chamber A, as indicated. When the water in the boiler is at its highest level the U-tube is filled with water

and the pump regulator valve V is not feeding. As the level of the water in the boiler drops the water recedes from the outer surface of the U-tube, and the upper branch of the tube is surrounded with steam.

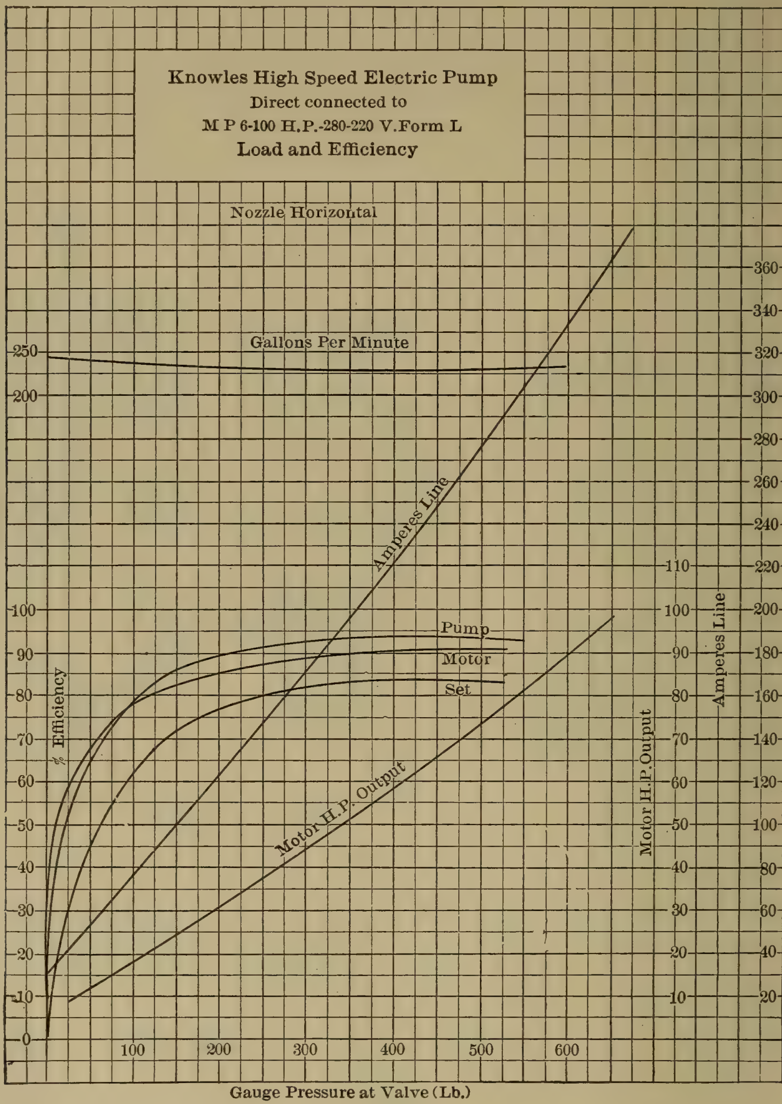


FIG. 390.

The steam causes the water in the vessel A to boil, and the pressure generated is transmitted through pipe F to diaphragm I, thereby opening controlling valve K. Wheel J permits of hand control. Regulators

of this type installed in the power plant of the Armour Institute of Technology are giving excellent service.

313. Power Pumps. — Piston pumps, geared, belted, or direct connected to electric motors, gas engines, and water motors, are used

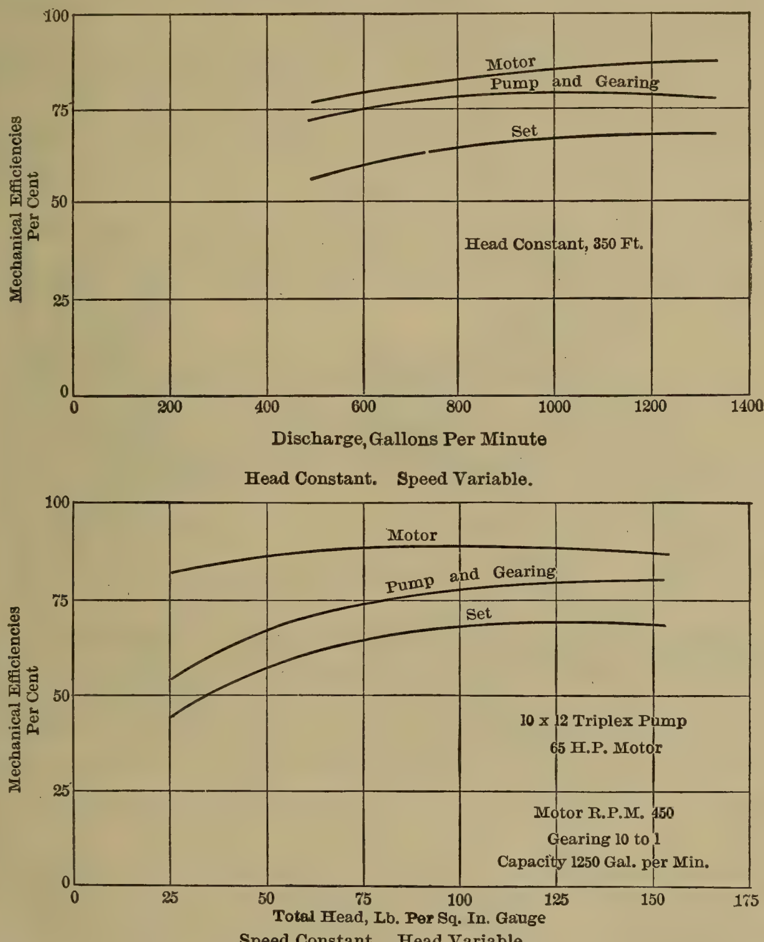


Fig. 391. Performance of a 65-horse-power, Motor-driven Triplex Pump. Geared Type.

chiefly where steam power is not available. Their general utility is evidenced by the rapidly increasing number installed in situations formerly occupied by the direct-acting steam pump. The efficiency of this type of pump depends in a large measure upon the character of the driving motor and the efficiency of the transmitting mechanism. High-speed power pumps direct connected to electric motors give

efficiencies from line to water horse power as high as 83 per cent, while the low-speed geared type seldom exceed 70 per cent. The curves in Fig. 390 give the performance of a direct-connected triplex pump, and those in Fig. 391 the performance of a triplex pump geared to an electric motor. Both of these performances are exceptionally good and are considerably above the average.

For a General Treatise on the Design and Operation of Pumping Machinery consult "Pumping Machinery," by A. M. Greene; John Wiley & Sons, 1911.

314. Injectors. — As a boiler feeder the injector is an efficient and convenient device, cheap and compact, with no moving parts, delivers hot water to the boiler without preheating, and has no exhaust steam to be disposed of. Its adoption in locomotives is practically universal, but in stationary practice it is limited to small boilers or single boilers or as a reserve feeder in connection with pumps. The objections to

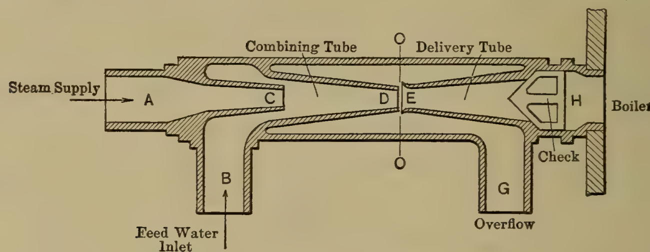


Fig. 392. Elementary Steam Injector.

an injector are its inability to handle hot water, the difficulty of maintaining a continuous flow under extreme variation of load, and the uncertainty of operation under certain conditions. Fig. 392 illustrates the simplest form of single-tube injector. Boiler steam is admitted at *A* and, flowing through nozzle and combining tube to the atmosphere through *G*, partially exhausts the air from pipe *B*, thereby causing the water to rise until it comes in contact with the steam. The steam emerging from nozzle *C* at high velocity condenses on meeting the water and imparts considerable momentum to it. The energy in the rapidly moving mass is sufficient to carry it across opening *O*, lift check *H* from its seat and force it into the boiler. The steam then ceases to escape at *G*.

315. Positive Injectors. — Fig. 393 shows a section through a Hancock injector, illustrating the principles of the double-tube positive type. Its operation is as follows: Overflow valves *D* and *F* are opened and steam is admitted, which at first passes freely through the overflow to the atmosphere and in so doing exhausts the air from the suction pipe. This causes the feed water to rise until it meets the jet of steam

and the two are forced through the overflow. As soon as water appears at the overflow, valve *D* is closed, valve *C* partially opened, and valve *F* closed. This admits steam through the forcing jet *W* and, the overflow valves being closed, the water is fed into the boiler. In case the action is interrupted for any reason it is necessary to restart it by hand.

The chief advantage of the double-tube positive type lies in its ability to lift water to a greater height and to handle hotter water than the single-tube. Its range in pressure is also greater, that is, it will start with a lower steam pressure and discharge against a higher back pressure. Double-tube injectors are used almost exclusively in locomotive work.

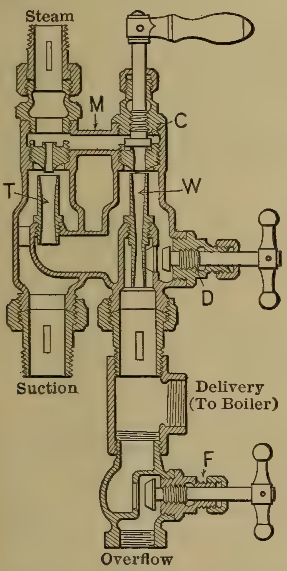


FIG. 393. Hancock Double-tube Injector.

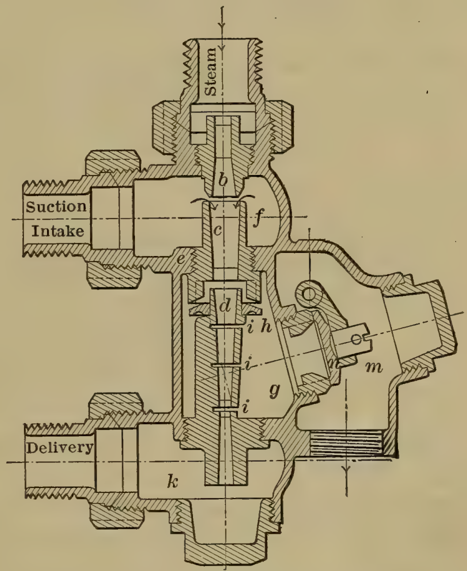


FIG. 394. Penberthy Automatic Injector.

316. Automatic Injectors. — Fig. 394 shows a section through the Penberthy injector. Its operation is as follows: Steam enters at the top connection and blows through suction tube *c* into the combining tube *d* and into chamber *g*, from which it passes through overflow valve *n* to the overflow *m*. When water is drawn in from the suction intake and begins to discharge at the overflow, the resulting condensation of the steam creates a partial vacuum above the movable ring *h* and the latter is forced against the end of tube *c*, cutting off the direct flow of water to the overflow. The water then passes into the boiler. Spill holes *i*, *i*, *i* are for the purpose of relieving the excess of water until

communication with the boiler has been established. The action of opening and closing the overflow is entirely automatic. Where the conditions are not too extreme the automatic injector is to be preferred for stationary work because of its restarting features. It is also used on traction, logging, and road engines, where its certainty of action and special adaptability render it invaluable for the rough work to which such machines are subjected.

Injectors, Theory of: Trans. A.S.M.E., 10–339; Sibley Jour., Dec., 1897, p. 101; Power, May, 1901, p. 23; Thermodynamics of the Steam Engine, Peabody, Chap. IX; Theory of the Steam Injector, Kneass.

Injectors, General Description: Engr. U. S., Oct. 1, 1907, Nov. 15, 1907, July 15, 1904, p. 501, Feb. 2, 1903, p. 151; Power, Aug., 1906, p. 478; Engr., Lond., March 10, 1905, p. 244; Engineering, Aug. 30, 1895, p. 281.

317. Performance of Injectors. — The performance of an injector may be very closely determined from the equation

$$w = \frac{xr + q - t + 32}{t - t_0} \quad \text{(Kneass, "Theory of the Injector," p. 83),} \qquad (240)$$

in which

w = pounds of water delivered per pound of steam supplied,
x = quality of the steam supplied,
r = heat of vaporization,
q = heat of the liquid,
t = temperature of the discharge water,
t_0 = temperature of the suction water.

Figs. 395, 396, and 397 give the performance of a Desmond automatic injector as tested at the Armour Institute of Technology. The results check very closely with those calculated from above equation. Referring to Fig. 395 it will be seen that the weight of water delivered per pound of steam decreases as the initial pressure is increased, all other factors remaining the same. From Fig. 396 it will be noted that the weight of water delivered per pound of steam decreases as the temperature of suction supply is increased up to a point where the injector "breaks" or becomes inoperative. This critical temperature varies with the different types of injectors, being highest for the double-tube type, but seldom exceeds 160 degrees F. Fig. 397 shows that the weight of water delivered per pound of steam is practically constant for all discharge pressures within the limits of the apparatus.

Table 99 gives the range of working steam pressures for standard "Metropolitan" injectors with varying suction heads and temperatures, and, though strictly applicable to this particular type only, is characteristic of all makes.

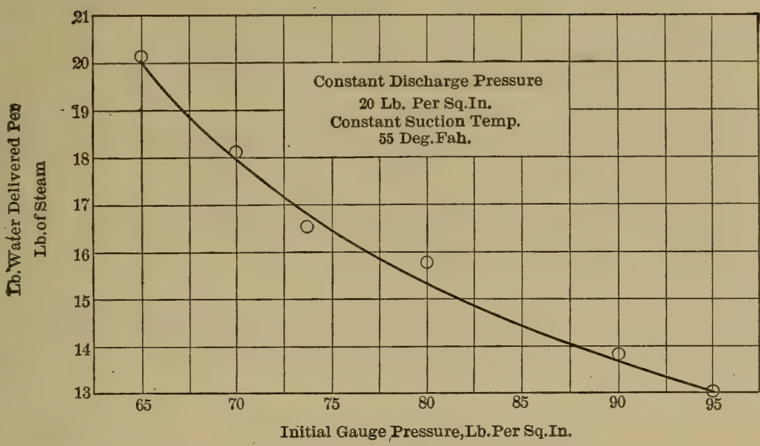


FIG. 395. Performance of an Automatic Injector with Varying Initial Pressure.

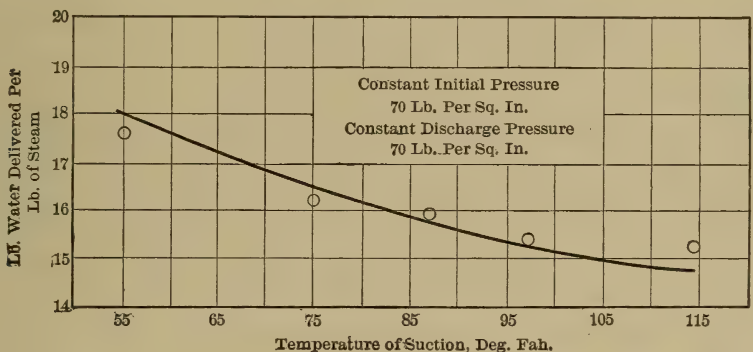


FIG. 396. Performance of an Automatic Injector with Varying Suction Temperature.

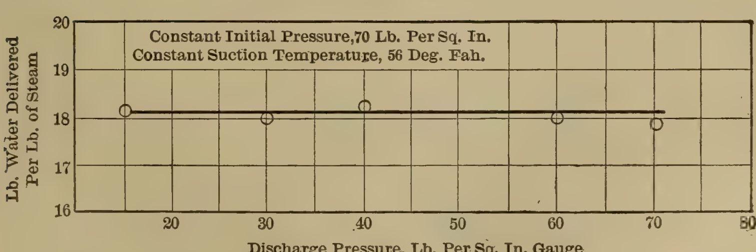


FIG. 397. Performance of an Automatic Injector with Varying Discharge Pressure.

In selecting an injector the following information is desirable for best results:

1. The lowest and highest steam pressure carried.
2. The temperature of the water supply.
3. The source of water supply, whether the injector is used as a lifter or non-lifter.
4. The general service, such as character of the water used, whether the injector is subject to severe jars, etc.

Injectors, Tests of: Eng. News, March 17, 1898, July 16, 1896, p. 39; Locomotive Engineering, May, 1900, p. 204; Power, Oct., 1904, p. 602; Railroad Gazette, Dec. 11, 1896; Thermodynamics of the Steam Engine, Peabody, Chapter IX; Theory of the Injector, Kneass.

TABLE 99.

RANGE IN WORKING PRESSURES.

Standard " Metropolitan " Steam Injectors.

Suction Temperature, Degrees F.	Automatic.				
	Suction Head, Feet.				
	2	8	14	20	Under Pressure.
Under 60	25 to 150	30 to 130	42 to 110	55 to 85	20 to 160
100	26 to 120	33 to 100	55 to 80	....	25 to 125
120	....	....	....	....	26 to 85
140	....	....	....	....	....

Suction Temperature, Degrees F.	Double Tube.				
	Suction Head, Feet.				
	2	8	14	20	Under Pressure.
Under 60	14 to 250	23 to 220	27 to 175	42 to 135	14 to 250
100	15 to 210	26 to 160	37 to 120	46 to 70	15 to 210
120	20 to 185	30 to 120	42 to 75	....	20 to 185
140	20 to 120	35 to 70	....	....	20 to 120

318. Injector *vs.* Steam Pump as a Boiler Feeder. — From a purely thermodynamic standpoint the efficiency of an injector is nearly perfect, since the heat drawn from the boiler is returned to the boiler again, less a slight radiation loss. As a pump, however, the injector is very inefficient and requires more fuel for its operation than very wasteful feed pumps. This is best illustrated by an example: An injector of modern construction will deliver say 15 pounds of water to the boiler per pound of steam supplied, with delivery temperature of 150 degrees F. This corresponds to a heat consumption of 71.3 B.t.u. per pound of water delivered, thus:

With initial pressure of 115 pounds absolute,

$$H = 1188.8.$$

Heat in the water delivered to the boiler,

$$150 - 32 = 118 \text{ B.t.u. above 32 degrees F.}$$

Heat of 1 pound of steam above a feed temperature of 150 degrees F.,

$$1188.8 - 118 = 1070.8 \text{ B.t.u.}$$

Heat required to deliver 1 pound of water to the boiler,

$$\frac{1070.8}{15} = 71.3 \text{ B.t.u.}$$

A simple direct-acting duplex pump consumes say 200 pounds steam per i.h.p. hour. Assume the extreme case where the exhaust steam will not be used for heating the feed water and the latter is fed into the boiler at 60 degrees F.

The heat supplied to the pump per i.h.p. hour,

$$200 \{1188.8 - (60 - 32)\} = 232{,}160 \text{ B.t.u.}$$

Assuming the low mechanical efficiency of 50 per cent, the heat required to develop one horse power at the water end will be

$$232{,}160 \div 0.50 = 464{,}320 \text{ B.t.u. per hour.}$$

Since the steam pressure is 100 pounds gauge, the equivalent head of water at 60 degrees F. is

$$2.3 \times 100 = 230 \text{ feet.}$$

Assume the friction in the feed pipe, the resistance of valves, etc., to be 30 per cent of the boiler pressure; the total head pumped against will be

$$230 + 69 = 299, \text{ say } 300 \text{ feet,}$$

$$1 \text{ horse-power hour} = 1{,}980{,}000 \text{ foot-pounds per hour,}$$

$$\frac{1{,}980{,}000}{300} = 6600 \text{ pounds;}$$

that is, 1 horse power at the pump will deliver 6600 pounds of water per hour to the boiler against a head of 300 feet.

The heat consumption per pound of water delivered,

$$\frac{464,320}{6600} = 70.3 \text{ B.t.u.}$$

If the feed water is heated to say 210 degrees F. by the exhaust steam from the pump, the heat consumption will be 63.7 B.t.u. as against 70.3 without the heater.

Thus even in this extreme case of poor steam-pump performance the heat consumption lies in favor of the pump. With the better grades of pumps this disparity is considerably greater, and decidedly so if the exhaust steam is used to preheat the feed water. For intermittent operation the condensation losses in the pump may more than offset this gain. Other conditions, however, such as compactness, low first cost, and ease of operation are oftentimes considerations and the heat consumption is of minor importance.

319. Air Pumps. — Condenser air pumps may be divided into two classes:

1. Wet-air pumps and
2. Dry-air pumps.

The former handle both air and water and the latter air alone. Ordinary jet-condenser wet-air pumps handle simultaneously the circulating water, condensed steam, and entrained air, and are, in fact, a combination of circulating and vacuum pump. Surface-condenser wet-air pumps are the same in principle and design, but are smaller in size for a given main engine output, as they handle the condensed steam and air only.

Wet-air pumps may be driven by the main engine or independently and may be direct acting, Fig. 296, or flywheel driven, Fig. 312. The flywheel type may be steam, electric, or belt driven. Dry-air pumps are virtually air compressors, as their function is to compress air from the pressure existing in the condenser to that of the atmosphere. They are generally of the flywheel type.

320. Dean Air Pump. — Fig. 398 shows a section of the air cylinder of a Dean twin-cylinder wet-air pump as applied to a standard low-vacuum jet condenser. There are three sets of valves, the suction or foot valves *A*, *A*, the lifting or bucket valves *B*, *B*, and the head or discharge valves *C*, *C*. On the upward stroke of the piston or bucket a partial vacuum is formed in the chamber between the bucket and the lower head, causing the water and air in the bottom of the barrel to

lift the foot valves A, A from their seats and flow into the cylinder. On the downward stroke the foot valves A, A close and water and air are entrapped in chamber R between the lower head and the bucket. As the bucket descends, the pressure of air in the cylinder lifts the bucket valves B, B from their seats and permits the air and water to escape to the upper portion S of the cylinder between the head plate and the bucket. On the next upward stroke the water and air are forced through the discharge valves C, C into the hot well. This discharge of water and air from the top compartment is simultaneous with influx of water and air in the lower chamber.

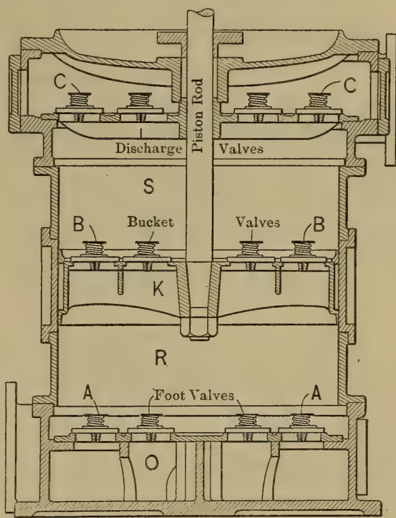


FIG. 398. Dean Air Pump.

See paragraph 242 for other types of wet-air pumps in connection with jet condensers.

321. Size of Wet-air Pumps; Jet Condensers. — In proportioning such pumps the quantity of cooling water and condensed steam to be taken care of is readily determined, but the percentage of air mingled with it must be estimated. Surface water under atmospheric pressure ordinarily contains from 2 to 5 per cent of air by volume. To provide for possible leakage a very liberal factor is usually allowed, an average figure being about 10 per cent.

Let Q = total volume of air and water in cubic feet per hour to be handled by the pump;
V = volume of cooling water in cubic feet per hour;
v = volume of condensed steam in cubic feet per hour;
v_a = volume of air at pressure p_a and temperature t_a;
t_a = temperature of the air entering the condenser, degrees F.;
t_2 = temperature of the discharge water, degrees F.;
t_o = initial temperature of the cooling water, degrees F.;
p_a = atmospheric pressure, pounds per square inch;
p_c = total pressure in the condenser, pounds per square inch;
p_v = pressure of aqueous vapor at temperature t_2;

then $(V + v)$ = volume of water to be pumped from the condenser per hour.

The air entering the condenser will be increased in volume on account of the reduction in pressure and the increase in temperature. If v_a is the original volume under pressure p_a and temperature t_a the final volume on entering the condenser is (see equations 163 and 164)

$$\text{Final volume} = v_a \frac{p_a}{p_c - p_v} \times \frac{t_2 + 460}{t_a + 460}, \tag{241}$$

and the total volume to be exhausted per hour by the pump is

$$Q = V + v + v_a \frac{p_a}{p_c - p_v} \times \frac{t_2 + 460}{t_a + 460}. \tag{242}$$

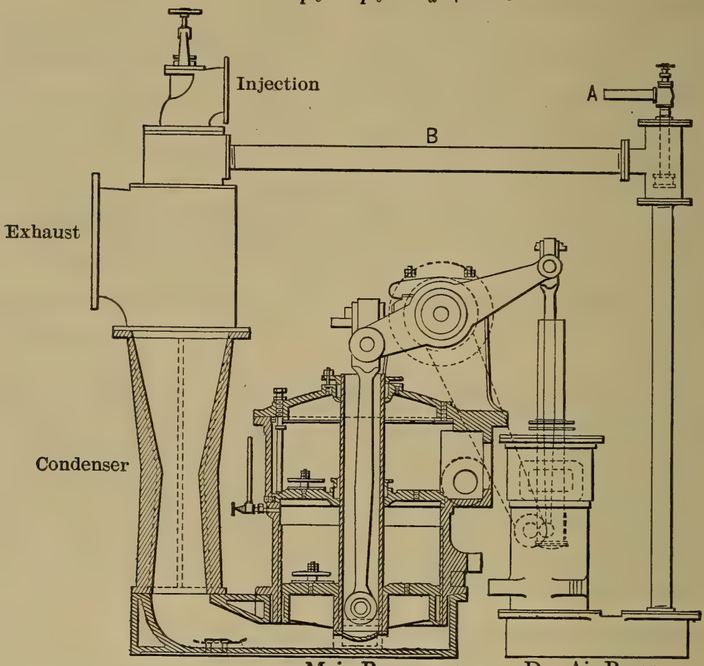


FIG. 399. Hewes and Phillips Jet Condenser and Air Pumps.

Under average conditions of reciprocating-engine practice the hot-well temperature is about 110 degrees F. and the absolute back pressure 4 inches of mercury. Assuming 70 degrees F. as the initial temperature of the circulating water and allowing 10 per cent as the air entrainment,

$p_a = 29.92$	$t_0 = 70$	$v = 0.04\,V$ (see equation 167)
$p_c = 4$	$t_2 = 110$	$v_a = 0.1\,V.$
$p_v = 2.59$	$t_a = t_0 = 70$	

Substitute these values in (242)

$$Q = V + 0.04\,V + 0.1\,V \frac{29.92}{4.0 - 2.59} \times \frac{110 + 460}{70 + 460}$$
$$= 3.3\,V.$$

Average practice gives 3 V as the pump displacement per hour for a single-acting pump and 3.5 V for a double-acting pump, the cylinders being ordinarily proportioned on a piston velocity of 50 feet per minute at rated capacity.

Table 100 gives the approximate sizes of air pumps for condensers as manufactured by prominent makers.

The combined air and circulating pump is not adapted for high vacuum work on account of the enormous increase in air volume at very low pressures. With cold injection water and a good air-tight condensing system vacua as high as 2 inches absolute are possible with the standard type of jet condenser air pumps but practice recommends the use of separate air and circulating pumps under these conditions. (See paragraph 326.)

TABLE 100.

APPROXIMATE SIZES OF PISTON AIR PUMPS FOR STANDARD LOW VACUUM CONDENSERS.

Pounds of Steam Condensed per Hour.	Jet Condenser.			Surface Condensers.	
	Duplex Pump.	Horizontal Double Acting Pump.	Vertical 2-Cylinder Single Acting.	Horizontal.	Vertical 2-Cylinder.
500 to 1,000	4¾ × 5	6 × 7	5 × 4	3½ × 4	....
1,000 to 1,500	5¾ × 6	8 × 7	6 × 4	4 × 4	....
1,500 to 2,000	6½ × 6	8 × 12	7 × 5	4 × 6	....
2,000 to 2,500	7½ × 6	9 × 9	9 × 6	5 × 7	....
2,500 to 3,000	7 × 10	9 × 10	10 × 8	5 × 8	....
3,500 to 4,000	8 × 10	11 × 12	11 × 9	6 × 8	....
4,000 to 4,500	8½ × 10	12 × 14	12 × 8	7 × 9	....
4,500 to 5,000	9 × 10	14 × 14	12 × 10	7 × 10	....
5,000 to 6,000	10 × 10	14 × 16	14 × 10	7 × 12	....
6,000 to 7,000	10½ × 10	15 × 16	15 × 10	8 × 10	8 × 4
7,000 to 8,000	11 × 10	15 × 18	15 × 12	8 × 12	8 × 6
8,000 to 9,000	12 × 10	16 × 18	16 × 10	9 × 12	9 × 6
9,000 to 10,000	12 × 15	18 × 18	17 × 12	10 × 12	10 × 8
10,000 to 15,000	15 × 15	20 × 24	20 × 12	12 × 14	11 × 8
15,000 to 20,000	17 × 15	24 × 24	22 × 15	14 × 16	12 × 8
20,000 to 25,000	19 × 15	26 × 24	24 × 18	16 × 24	14 × 10

Wet-air pumps are usually independently driven, making it possible to vary the speed of the pump irrespective of the engine speed and to create a vacuum before starting the engine. Occasionally, however, when the load is constant, as in pumping-engine practice, the pump may be driven by the main engine.

Centrifugal wet-air pumps are much in evidence in modern stations and offer many advantages over the piston type. See paragraphs 330 to 334.

322. Edwards Air Pump. — Fig. 400 shows a section through the air cylinder of an Edwards air pump. This device belongs to the surface-condenser "wet-air pump" class, as both the water of condensation and the entrained air are exhausted simultaneously by the same piston. Unlike the standard type of wet-air pumps, foot valves and bucket valves are entirely dispensed with. The condensed steam flows continuously by gravity from the condenser into the base of the pump through passage A and annular space B. As the piston C descends it forces the water from the lower part of the casing F into the cylinder proper through the ports P, P. On the upward stroke the ports in the piston are closed and the air and water discharged through head valves D and exhaust port E to the hot well. The seats of valves D are constructed with a rib between each valve and a lip around the outer edge, so that each valve is water-sealed independently of the others. In earlier air pumps of this general type the clearance between the bucket and head valve seat is necessarily large, due to the space occupied by the bucket valves and the ribs on the under side of the valve seating. This clearance space reduces the capacity of the pump, since the air above the bucket must be compressed above atmospheric pressure before it can be discharged, and on the return stroke will expand and occupy a space which should be available for a fresh supply of air from the condenser. In the Edwards air pump the clearance space is reduced to a minimum, since there are no bucket valves to limit it. The absence of suction or foot valves still further increases the capacity of the pump for similar reasons. These pumps are arranged either single, double, or triplex; steam, electric, or belt driven; slow or high speed. They are ordinarily used in connection with surface condensers.

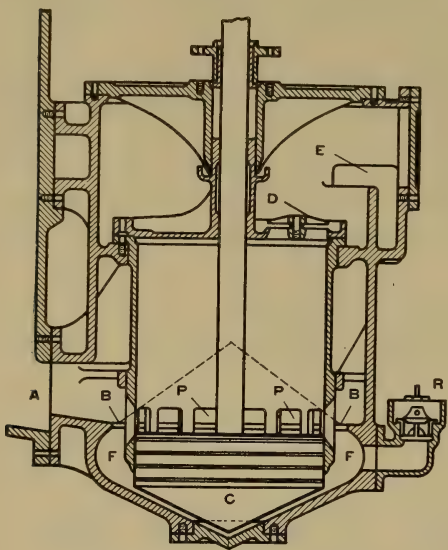


FIG. 400. Edwards Air Pump.

Centrifugal Wet Vacuum Pump: Power & Engr., Jan. 4, 1910.

323. Mullan Valveless Air Pump. — Fig 401 shows a section through the "Mullan valveless air pump" as used in connection with the C. H.

Wheeler Company's "high-vacuum" condensing outfit. The pump is double acting and devoid of suction valves. The cylinder has a central port which is uncovered by the piston at each end of the stroke and covered at all other positions. Discharge valves of the Gutermuth spiral-spring type are located in both heads of the cylinder. As the

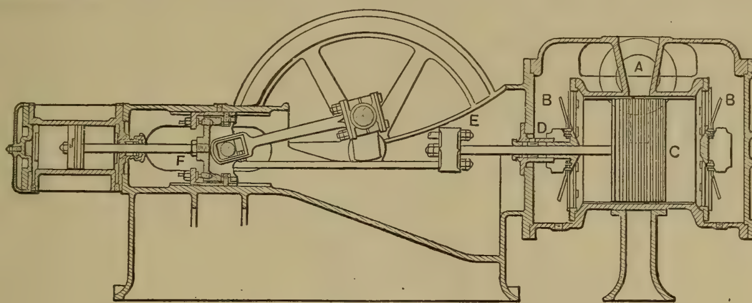


FIG. 401. Mullan Air Pump.

piston moves from one end of its stroke to the other it forms a vacuum behind it and forces out the gases and water ahead of it; when it reaches the end of the stroke the central inlet port is uncovered and the vacuum behind the piston draws in the condensation and gases from the condenser. This operation is repeated on the return stroke.

The makers claim that the pump will operate, under shop-test conditions, within one half inch of the barometer, enabling them to guarantee a vacuum within two inches of absolute under full-load conditions of steam turbine operation.

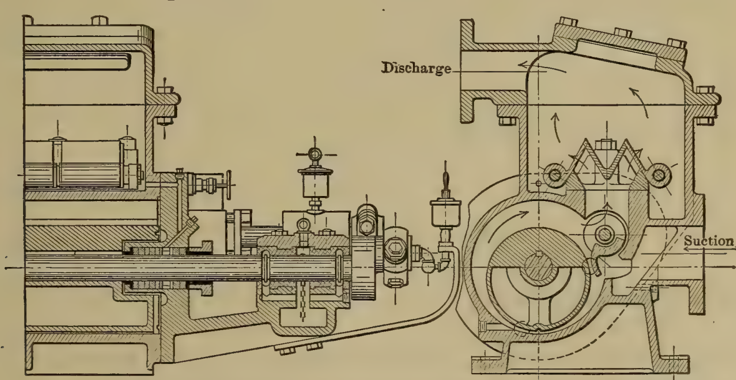


FIG. 402. High Vacuum "Rotrex" Pump.

324. C. H. Wheeler "Rotrex" Pump. — Fig. 402 shows a partial axial and an end section through a C. H. Wheeler & Co.'s high-vacuum "Rotrex" pump. This pump is of the wet-vacuum type and handles

both air and water of condensation but it is also adapted for dry-air purposes. The apparatus consists of a cylindrical casing and a rotor mounted eccentrically on the shaft. This shaft is carried in outboard ring oil bearings which are entirely independent of the stuffing boxes. The division between the suction and discharge space in the pump cylinder is maintained by a radius cam carried on a shaft independent of the stuffing boxes. This cam is operated from the rotor shaft by a lever and crank on the outside of the casing. The clearance spaces are water sealed. The discharge valves are of the Gutermuth type. Pump speed 200 to 300 r.p.m. The manufacturers guarantee that on dead-end test a vacuum may be obtained within one half inch of the barometer, and within one inch of the barometer under operating conditions.

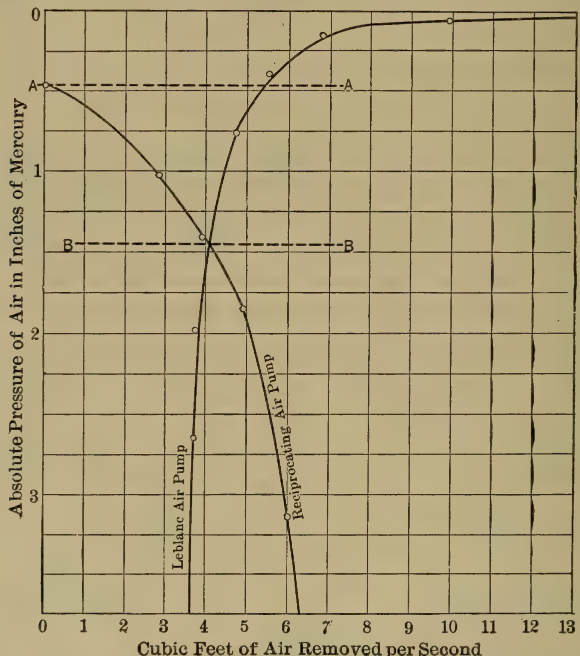


FIG. 403. Comparative Tests — Reciprocating Air Pump *vs.* Leblanc Air Pump.

325. Leblanc Air Pump. — The Leblanc air pump, described in paragraph 253, is finding much favor with engineers for high-vacuum service, and is supplanting the reciprocating type of pump to a considerable extent. The absence of valves, high rotative speed and the elimination of clearance spaces, enable large volumes of air to be handled at very low absolute back pressures. Since there are no reversals in operation no damage results from water being drawn into the air pump as when the hot-well pump fails to operate. For low vacua the recip-

rocating air pump is more effective, volumetrically, than the Leblanc pump, but for high vacua the latter gives the best results. This is shown by the curves in Fig. 403 which, though strictly applicable only to the conditions under which the tests were made, represent the general characteristics of the two types of pumps. (Jour. Wes. Soc. of

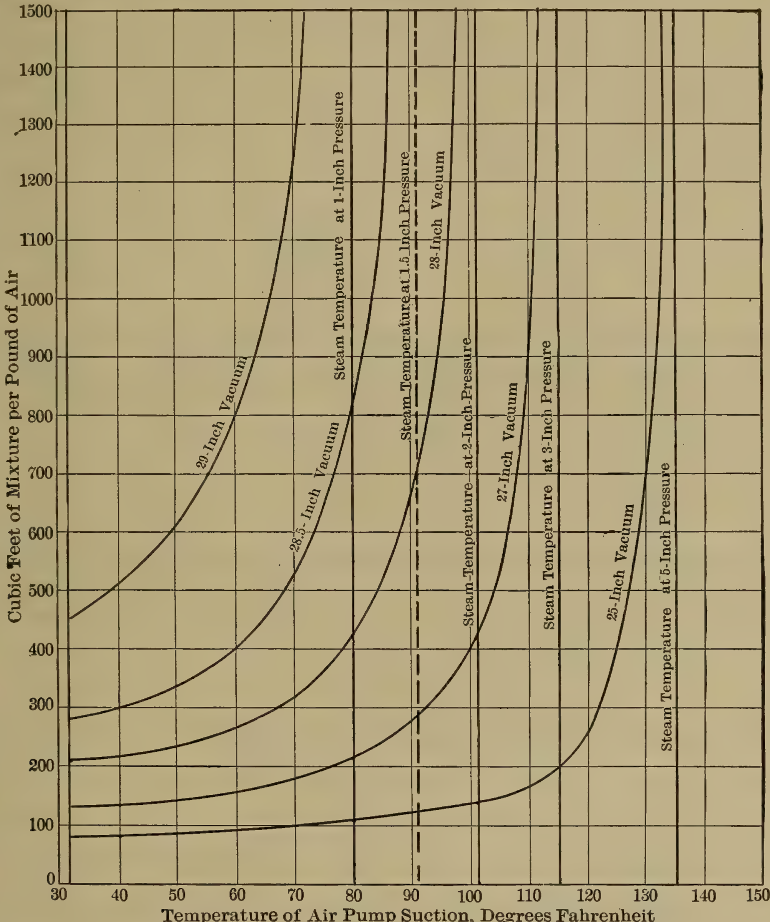


FIG. 404. Cubic Feet of Saturated Air Containing One Pound of Dry Air for Various Vacua and Air Temperatures.

Engrs., May, 1912, p. 430.) Referring to the curves, the reciprocating pump is superior to the Leblanc for vacua below the line *BB* and for vacua above *BB* the latter is the more effective. For vacua above line *AA* the Leblanc pump is in a class by itself. The power requirement of the Leblanc pump is considerably more than that of the reciprocating pump but with high vacua its volumetric capacity is so much greater

than the latter that the decreased air tension and corresponding increase in vacuum may more than offset power requirement. Table 101 gives the results of a Leblanc condenser test and gives some idea of the power requirements.

TABLE 101.

TEST OF LEBLANC CONDENSER AND PUMP.

Steam condensed, pounds per hour	13,100	21,100	28,750	37,000	44,400
Approximate rating, per cent	44	70	95	123	148
Ratio water to steam	82.5	51.4	45.8	33	24.9
Injection temperature, degrees F	65	70	71	70	70
Temperature difference between vacuum and discharge, degrees F	6	7	3.5	2	4
Vacuum, inches of mercury	28.76	28.09	27.96	27.59	26.81
Per cent of vacuum realized	99.1	98.6	99.16	99.15	98.43
Power to operate all pumps, e.h.p	52	53	56	55	51

326. Size of Wet-air Pump for Surface Condenser. — Since the wet-air pump for surface condenser handles only the condensed steam and air, its theoretical capacity, neglecting clearance, may be determined by eliminating V from equation (242) which then becomes

$$Q = V + V_a \frac{p_a}{p_c - p_v} \times \frac{t_2 + 460}{t_a + 460}. \tag{243}$$

The volume of air entering the condenser varies so much with the character of the power-plant equipment and the conditions of operation that any assumed average value of v may lead to serious error.

Average steam turbine practice gives

$$Q = 20\ v \text{ for 26-inch vacuum,}$$
$$Q = 30\ v \text{ for 27-inch vacuum,}$$
$$Q = 40\ v \text{ for 28-inch vacuum.}$$

Average reciprocating engine practice gives

$$Q = 85 \text{ per cent of above for same operating conditions.}$$

The air-pump displacement necessary to exhaust a given weight of air for different vacua and air-pump suction temperatures is shown in Fig. 404. The curves are based upon equation (241) and give the volume of *saturated* air containing one pound of *dry* air at various vacua and air-pump suction temperatures. The great reduction in volume effected by cooling the air-pump suction is also clearly shown. The marked superiority of the counter-current condenser over the parallel-flow condenser for high vacua is chiefly due to the reduction in temperature of the non-condensable vapors.

327. Alberger Rotative Dry-air Pump. — Fig. 405 shows a section through the air cylinder of an Alberger rotative dry-air pump, illustrating a type of pump in which the admission valve is mechanically operated. This pump is designed to operate with dry air only, all condensation being removed before the air enters the cylinder. This permits of the use of a small clearance space and makes it possible to run at a higher speed of rotation than can be secured with a type of pump in which water is used to seal the valves. Referring to Fig. 405, air is being taken into the right-hand end of the cylinder through inlet *A* and forced from the left-hand end through exhaust opening *B*. Rotary valve *O* mechanically opens to admission and mechanically

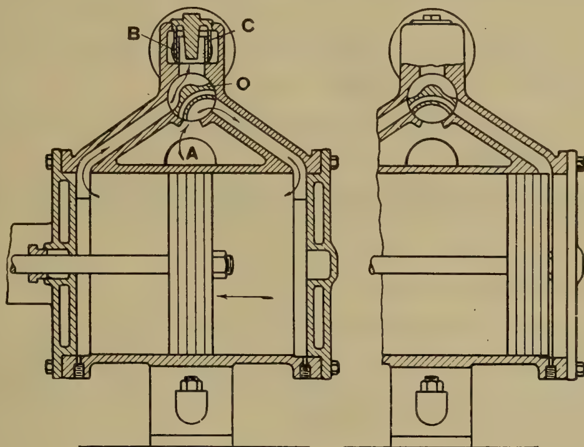


Fig. 405. Alberger Rotative Dry-air Pump.

closes the discharge. The discharge opening depends on the spring-regulated valve *C* at the top of the cylinder. Heads are water jacketed. Ports and passages are made large to reduce the friction of the air entering the pump, and, to obviate the bad effects of clearance, an equalizing passage is provided in valve *O*. The action of the passage is shown in the section to the right. When the piston reaches the end of the stroke the clearance space is filled with air at atmospheric pressure. If this pressure were not relieved the piston would travel a considerable distance before drawing in air from the condenser. By means of the equalizing passage the clearance space is connected to the opposite end of the cylinder and the vacuum there reduces the pressure in the clearance space.

328. Size of Dry-air Pumps. — "Dry-air" pumps are used in connection with barometric and surface condensers where a high degree of vacuum is essential, as in steam turbine practice. Such pumps are intended to exhaust the saturated non-condensable vapors only.

The capacity of the dry-air pump is based upon experience rather than theory. Current practice gives

$Q = 20\,v$ to $30\,v$ for vacua under 27 inches.

$Q = 35\,v$ to $50\,v$ for vacua of 28 inches and over, both referred to a 30-inch barometer.

Professor Weighton states that "with suitable condenser arrangements and a reasonably air-tight system there is nothing gained in efficiency by the use of air pumps exceeding in capacity 0.7 of a cubic foot per pound of steam condensed up to a vacuum of 29 inches." (Engineering Record, May 19, 1906, p. 61.)

The work done by the average "high-vacuum" reciprocating dry-air pump is a maximum for vacua between 18 and 20 inches.

This may be proved from Fig. 406 which represents a theoretical indicator card from the air-pump cylinder.

Let p_2 = pressure in the condenser, pounds per square inch absolute;
p_1 = atmospheric pressure;
v_2 = piston displacement, including clearance, cubic feet;
v_1 = volume of air in the cylinder when the valve opens to atmosphere, cubic feet;
v_c = clearance volume, cubic feet.

The work done is proportional to the area $ABCD$.

Area $ABCD$ = work done = area $EBIO + BAGI - FAGO - ECDF$.

Neglecting the exponential factor n for the sake of simplicity, thus making $pv = p_1v_1 = p_2v_2$ = constant,

$$W = \text{work done} = p_1v_1 + p_1v_1\int_{v_2}^{v_1}\frac{dv}{v} - p_2v_2 - p_1v_c + p_2v_c. \qquad (244)$$

Substitute p_1v_1 for its equivalent p_2v_2 and $\frac{p_1v_1v_c}{v_2}$ for its equivalent p_2v_c and integrate.

$$W = p_1v_1 + p_1v_1 \log_e v_2 - p_1v_1 \log_e v_1 - p_1v_1 + \frac{p_1'v_1v_c}{v_2}, \qquad (245)$$

making the first derivative zero.

$$\frac{dw}{dv} = 0 = \log_e v_2 - 1 - \log_e v_1 + \frac{v_c}{v_2}, \qquad (246)$$

$$0 = \log_e \frac{v_2}{v_1} - 1 + \frac{v_c}{v_2}, \qquad (247)$$

i.e., W is a maximum when

$$-\log_e \frac{v_2}{v_1} = \frac{v_c}{v_2} - 1, \qquad (248)$$

or

$$\log_e \frac{p_1}{p_2} = 1 - \frac{v_c}{v_2}, \text{ since } p_1v_1 = p_2v_2. \qquad (249)$$

For average high-vacuum practice $v_c = 3$ per cent of the piston displacement. Assume $v_2 = 1$, $v_c = 0.03$, and $p_1 = 14.7$ pounds per square inch, and substitute these values in Equation (249), thus:

$$\log_e \frac{14.7}{p_2} = 1 - \frac{0.03}{1.00}.$$

Whence $p_2 = 5.5$ pounds per square inch absolute, which corresponds to a vacuum of 18.6 inches of mercury.

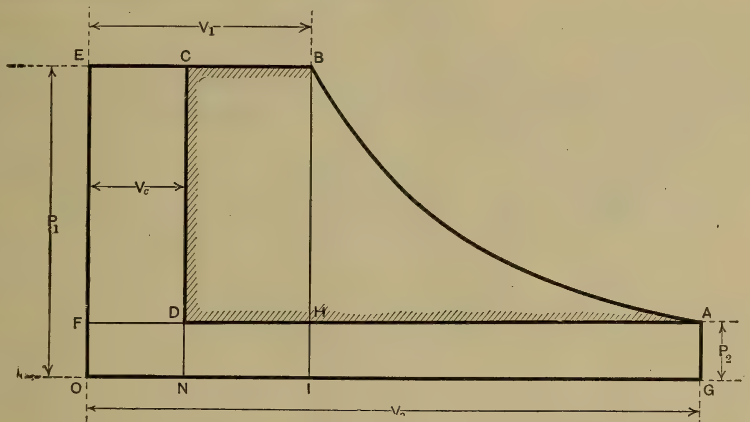


FIG. 406.

Thus we see that the maximum load on the pump occurs when the vacuum is between 18 and 20 inches. If the vacuum is less than this, the load falls off because of the decreased difference in pressure. If the vacuum is greater, the load falls from the decrease in weight of air handled.

329. Centrifugal Pumps. — Centrifugal pumps consist of two essential elements, (1) a rotary impeller which draws in the water at its center and (2) a stationary casing which guides the water thrown from the ends of the impeller to the discharge outlet. Increase of peripheral speed increases the energy in the impeller. This increase in energy may take the form of increase in pressure or potential energy, or it may be in the form of increase in rate of flow or kinetic energy. In general there is an increase in both kinetic and potential energy. The impeller may be of the open type, Fig. 408 (B), or closed, Fig. 408 (A). The casing may be cylindrical and concentric with the impeller, Fig. 412, or of spiral form, Fig. 407. It may be plain or fitted with diffusion vanes and any number of impellers may be employed. The shape of the impeller and casing and the number of impellers or stages determine the efficiency of the pump and its adaptability to certain conditions of service.

Centrifugal pumps are generally classified as

1. Volute
2. Turbine.

330. Volute Pumps. — Fig. 407 gives an end view of a typical single-stage volute pump with end plate removed so as to expose the impeller, and Fig. 409 shows a section through a modern single-stage volute pump with double suction. In the volute pump the casing is of spiral design forming a gradually increasing water or "whirlpool" chamber, *A–B*, Fig. 409, for the purpose of partially converting velocity head to pressure head. The older forms of volute pumps were very inefficient, seldom delivering more than 40 per cent of the energy supplied and usually not adapted to lifts greater than 50 feet. The modern pumps give efficiencies as high as 80 per cent, and the lift is limited only by the speed of the impeller. As a general rule the volute pump is of single-stage construction and limited to comparatively low lifts, 120 feet and under, though two-stage pumps of this type are on the market designed for heads as high as 1000 feet.

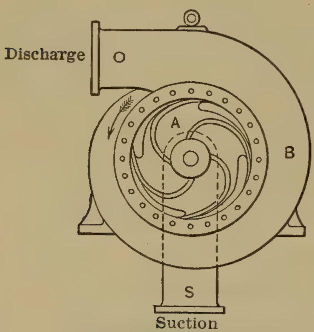


FIG. 407. A Typical Centrifugal Pump.

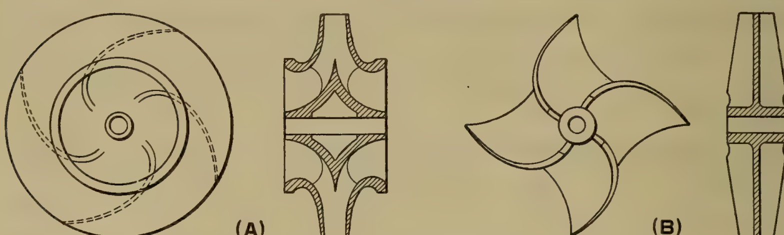


FIG. 408. Types of Impellers.

331. Turbine Pumps. — In the usual design of volute pumps the stream of water in the casing is at cross current with that thrown out from the impeller as shown in Fig. 410. The turbine pump is provided with a system of diffusion vanes or expanding ducts, disposed between the periphery of the impeller and the annular casing, somewhat like the guide vanes in a reaction turbine water wheel, so that the fluid emerges tangentially at about the velocity in the casing (see Fig. 411). The casing is usually concentric with the impeller and of uniform cross section though the volute casing is sometimes used in this connection. For high lifts these pumps are compounded, thereby reducing the

peripheral velocity and decreasing the friction losses. Fig. 412 shows a section through a three-stage Worthington turbine pump as installed

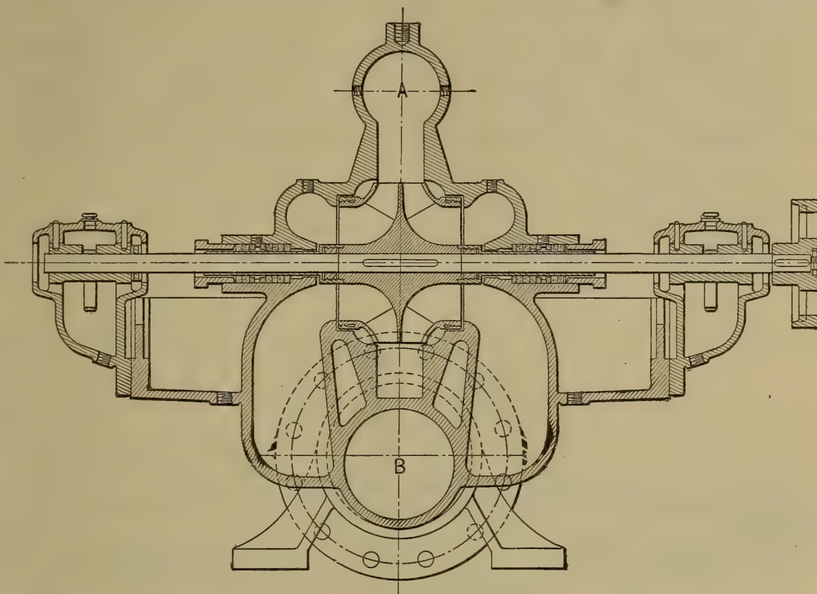


Fig. 409. Typical Single-stage Double Suction Volute Pump.

in the testing laboratories of the Armour Institute of Technology and designed to deliver 200 gallons per minute against a 750-foot head at 2500 r.p.m.

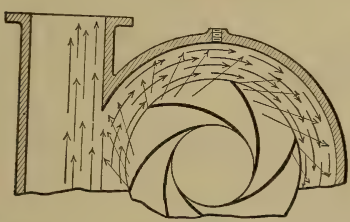


Fig. 410. Direction of Water from the Impellers of a Centrifugal Pump without Diffusion Vanes.

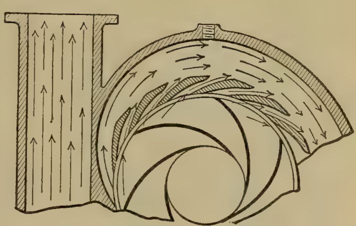


Fig. 411. Effect of Diffusion Vanes on the Direction of Water.

332. Field for Centrifugal Pumps. — In view of past developments it is probable that the centriugal pump will supplant the piston type of pump for practically all purposes, except perhaps for deep-well service and for very heavy pressures. Centrifugal pumps are now used for boiler feeding, circulating condensing water, hot-well and wet-vacuum purposes and for various applications of industrial service. Efficiencies

above 70 per cent are not unusual and the head against which the pump may operate is limited only by the peripheral speed at which the impeller may be safely run. Although the equivalent heat efficiency of the high-grade piston pump is superior to that of the centrifugal pump, other items, such as low first cost, decreased cost of repairs and the like, frequently offset this advantage. Some of the advantages of the centrifugal pump as compared with the piston type are:

1. Low first cost,
2. Compactness,
3. Absence of valves and pistons,
4. Low rate of depreciation,

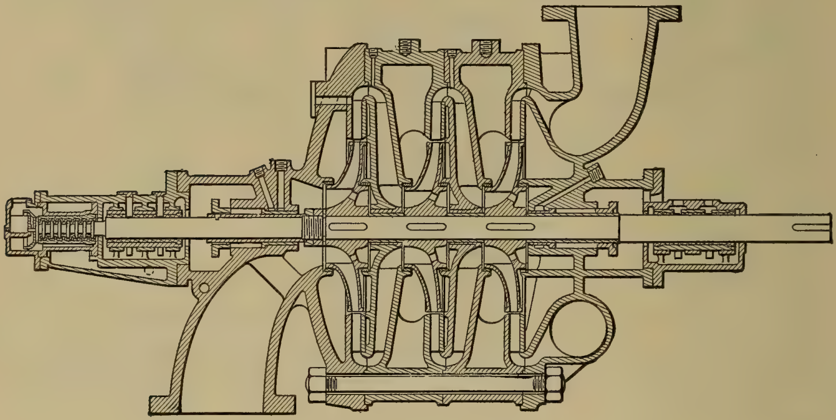


FIG. 412. Worthington Three-stage Turbine Pump.

5. Uniform pressure and flow of water,
6. Simplicity of design and ease of operation,
7. Freedom from shock,
8. High rotative speed, permitting direct connection to electric motors and steam turbines,
9. Ability to handle dirty water, sewage and the like,
10. In case of stoppage of delivery, the pressure cannot increase beyond the predetermined working pressure, and
11. Ease of repair.

Some of the disadvantages are:

1. Efficiency not as high as the best grade of piston pumps,
2. Cannot be direct connected to low-speed engines when high lifts are desired, and
3. The rate of flow cannot be efficiently regulated for wide ranges in duty.

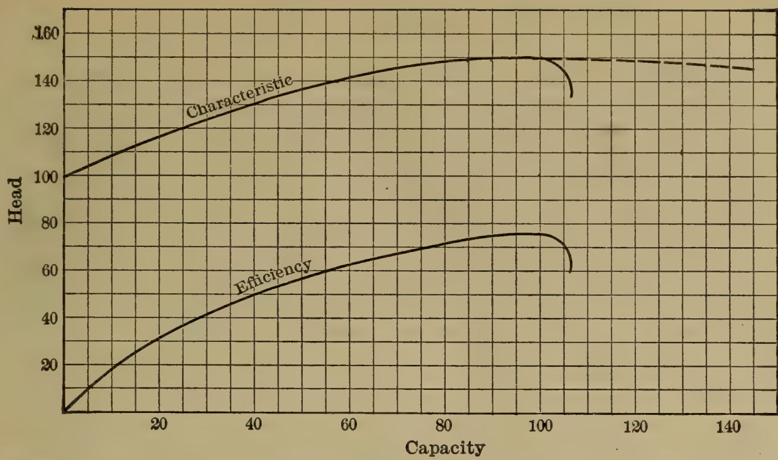


Fig. 413. Centrifugal Pump Characteristic for Hydraulic Elevator Service, Boiler Feeding, etc.

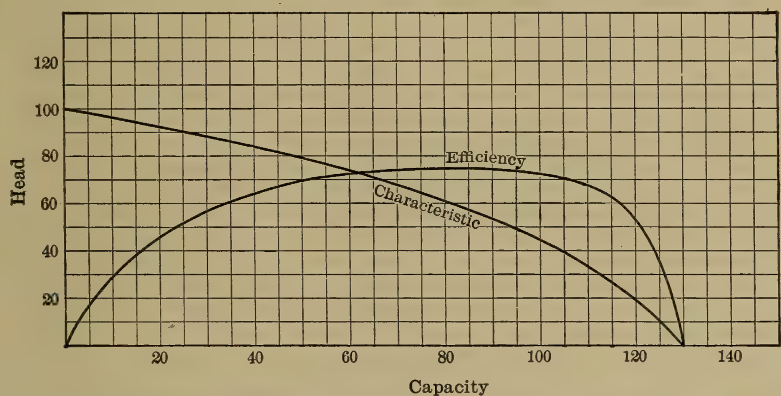


Fig. 414. Centrifugal Pump Characteristic for Dry-Dock Service.

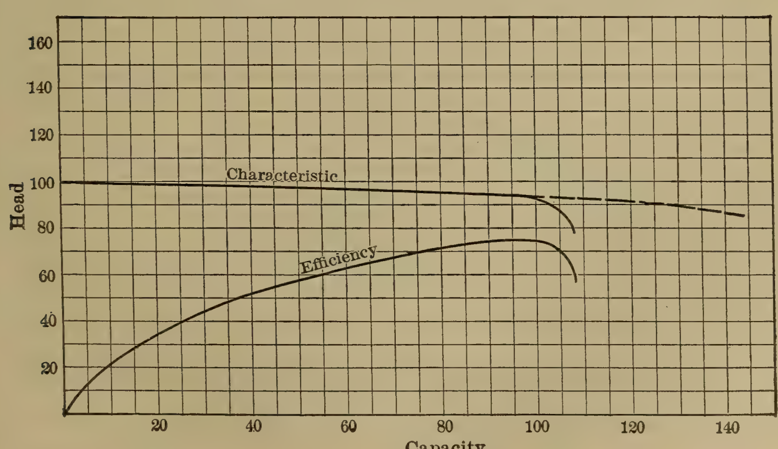


Fig. 415. Centrifugal Pump Characteristic for Water Works with Large Friction Head.

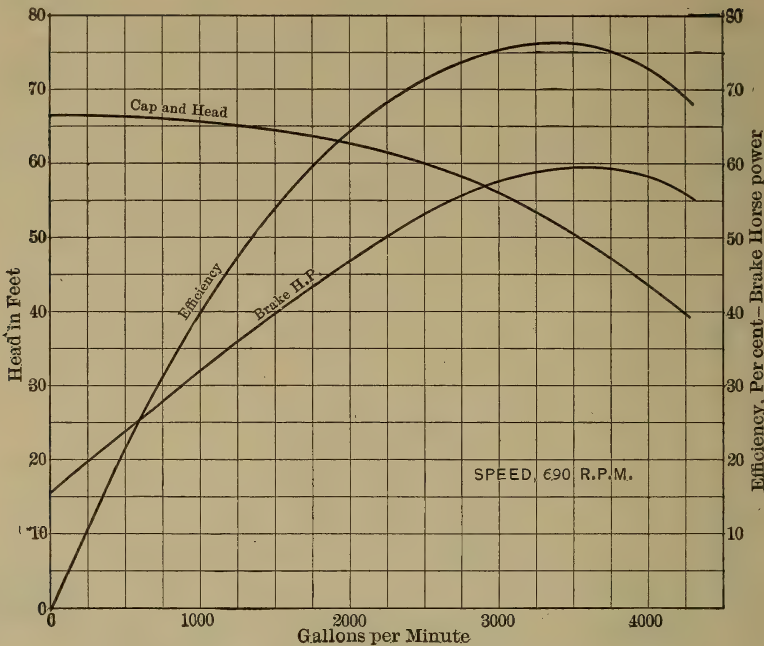


FIG. 416. Performance of Worthington 10-inch Volute Pump.

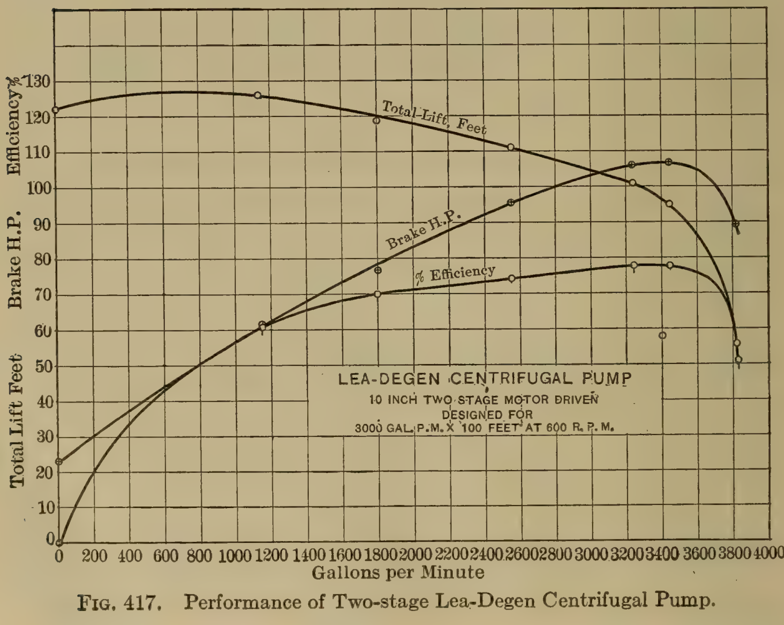


FIG. 417. Performance of Two-stage Lea-Degen Centrifugal Pump.

333. Performance of Centrifugal Pumps. — For best efficiency a centrifugal pump must be properly designed for the intended service as to curvature of vanes, diameter and speed of impeller, and number of stages. Figs. 413 to 415 are based upon experiments with De Laval centrifugal pumps. When a practically uniform head is required at constant speed with varying water supply as in city water works, hydraulic elevator systems or boiler feeding, the impeller vanes are designed to give the characteristic curve illustrated in Fig. 413 which protects the motor from possible overload.

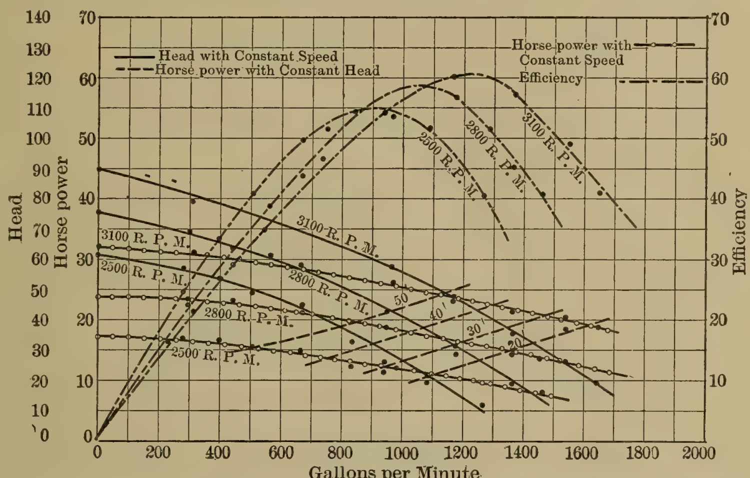


FIG. 418. Characteristic Curves of 8-inch Kerr Centrifugal Pump for Low Heads and Steam Turbine Speeds.

In dry-dock and other variable-head work, in order not to overload the motor, the power should be practically constant through wide variations of head and at the same time the efficiency should not vary seriously. A desirable characteristic for such a pump is illustrated in Fig. 414.

In water-supply systems in which the friction of the piping is a large part of the total head at full delivery, the characteristic shown in Fig. 415 is especially useful. Thus, when the system reduces its demand for water and the frictional head is consequently considerably reduced, the pump would automatically adjust itself to the reduced head without change of speed. Figs. 416 to 419 are based upon experiment and show the relationship between speed, head, capacity, efficiency and power consumption of various types of pumps.

The curves in Fig. 418 are of interest in that they show the characteristic of a centrifugal pump operating at low heads for very high rotative speed.

TABLE 102.

DATA PERTAINING TO SINGLE-STAGE CENTRIFUGAL PUMPS — CLASS "A," LAWRENCE PUMPS

Diameter Discharge, Inches.	Diameter Suction, Inches.	Diameter Impeller, Inches.	Rated Capacity, Gals. per Min.	Velocity of Discharge, Ft. per Min.	Horse Power for Each Foot of Lift.	Total Head in Feet and Revolutions per Minute.										
						16 Ft.	20 Ft.	25 Ft.	30 Ft.	35 Ft.	40 Ft.	50 Ft.	60 Ft.	70 Ft.	80 Ft.	90 Ft.
1	1½	11	25	120	0.025	760	850	950	1040	1120	1200	1340	1470	1590	1695	1800
1½	2	14	55	120	0.042	620	690	765	830	890	950	1050	1140	1230	1315	1400
2	2½	18	100	120	0.063	460	510	570	620	670	710	795	870	940	1000	1060
3	3½	22	242	132	0.136	380	420	470	510	550	580	645	700	750	800	850
4	4½	26	430	132	0.217	310	340	370	405	435	465	515	560	600	640	670
5	6	29	734	144	0.309	279	295	320	350	370	400	435	470	510	545	570
6	6	32	1050	144	0.446	240	265	290	320	340	360	395	435	465	490	520
7	7	34	1439	144	0.606	220	250	275	300	320	340	375	405	435	465	490
8	8	36	1880	144	0.791	210	235	260	280	300	320	355	390	415	440	465
10	10	42	2938	144	1.265	190	210	230	247	265	280	310	335	360	380	400
12	12	45	4230	144	1.765	175	195	215	230	247	265	290	315	335	358	377
15	15	48	6610	144	2.775	165	185	200	220	235	250	275	295	317	335	355
18	18	52	9378	144	4.000	150	170	185	200	215	228	250	270	290	310	325

Tables 102 to 104 give the capacity, speed, head and power requirements for commercial sizes of centrifugal pumps, and may be used as a guide in selecting the size of pump for general service.

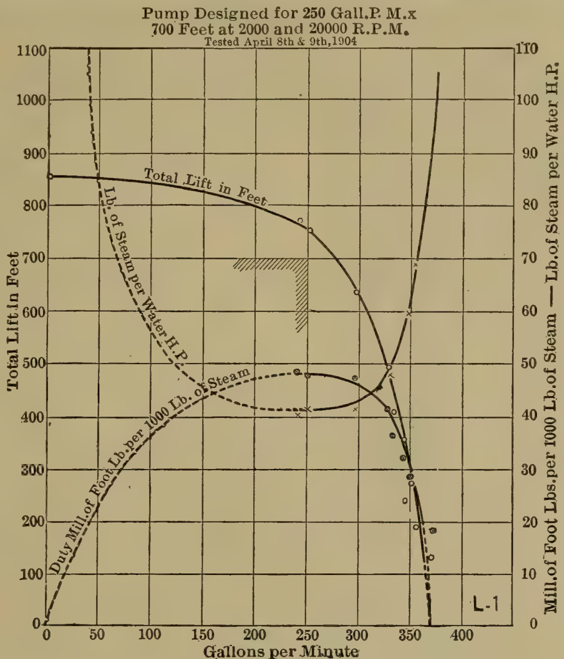


FIG. 419.

TABLE 103.

DATA PERTAINING TO WORTHINGTON MULTI-STAGE TURBINE PUMPS.

Diameter of Discharge Pipe, Inches.	Diameter of Suction Pipe, Inches.	Rated Capacity, Gallons per Minute.	* Horse Power per Foot of Lift.	Total Head in Feet, R.P.M., Number of Stages. 100 Ft. R.P.M.	100 Ft. Number of Stages.	200 Ft. R.P.M.	200 Ft. Number of Stages.	300 Ft. R.P.M.	300 Ft. Number of Stages.	400 Ft. R.P.M.	400 Ft. Number of Stages.
1	1.5	30	0.02	2000	2	......	....	......	....	......	......
1.5	2	45–60	0.0395	1500	2	2000	3	......	....	......	......
2	2.5	75–100	0.0625	1300	2	1800	3	1500	4	......	......
2.5	3	125–150	0.095	1200	2	1600	3	1300	4	......	5
3	4	200–250	0.134	1100	2	1400	3	1200	4	......	5
4	5	350–450	0.222	950	2	1200	3	1000	4	......	5
5	6	600–700	0.297	800	1	1300	2	1150	3	1050	4
6	7	800–1000	0.396	750	1	1200	2	1000	3	950	4
8	10	1500–1800	0.643	600	1	1000	2	800	3	780	4
10	12	2500–2800	1.00	500	1	800	2	700	3	670	4

* Horse power based on maximum capacity.

TABLE 104.

CAPACITIES, HEADS AND SPEEDS OF McEWEN BROS. PUMPS

Double Suction Hot-well Pumps.

Size.	Speed, R.p.m.	Range of			
		Economical Capacities, Gal. per Min.	Economical Head.	Brake H.p.	Pipe Velocities, Ft. per Sec.
In.			Ft.		
	3600	125– 160	60–50	4.5– 4.7	5.7– 7.2
3	3200	110– 145	45–35	3.2– 3.5	5.0– 6.1
	2800	100– 120	35–30	2.4– 2.5	4.5– 5.4
	3600	250– 325	60–50	8.1– 8.75	6.4– 8.3
4	3200	225– 300	45–35	5.7– 6.2	5.8– 7.7
	2800	200– 250	35–30	4.1– 4.3	5.1– 6.4
	3600	450– 600	85–70	20 –22	7.4– 9.8
5	3200	400– 550	65–55	14 –16	6.6– 9.0
	2800	350– 500	50–40	9.7–11	5.8– 8.2
	3600	700–1000	85–70	30 –36	8 –11.4
6	3200	650– 900	65–55	23 –27	7.4–10.2
	2800	600– 800	50–40	16 –18	6.8– 9.1

Turbine Pumps: Cassier's Mag., April 12, 1911; Engng., Jan. 26, 1912; Eng. Rec., June 15, 1912.

334. Rotary Pumps. — Rotary pumps are often used for circulating cooling water in condenser installations, and give about the same efficiency as centrifugal pumps under similar conditions of operation. For moderate pressure and large volumes they offer the advantage of low rotative speed, thus permitting direct connection to slow-speed steam engines. At high speeds they are noisy, due chiefly to the gearing. They occupy considerably less space than piston pumps of the same capacity, but require more room than the centrifugal type.

Fig. 420 shows a section through a two-lobe cycloidal pump. The shafts are connected by wheel gearing, the power being applied to one of the shafts. The water is drawn in at I and forced out at O, the displacement per revolution being equal to four times the volume of chamber A. There is no rubbing between impellers and casing. In this type of pump the pressure is independent of the speed of rotation, and the capacity varies almost directly with the speed. The slip varies from 5 to 20 per cent according to the discharge pressure.

Fig. 421 shows a section through a rotary pump with movable butment. Fig. 422 illustrates the performance of a 45-mm. Siemens-Schuckert rotary pump at different speeds and discharge pressures. (Zeit. d. Ver. Deut. Ing., June 24, 1905, p. 1040.) Large rotary pumps

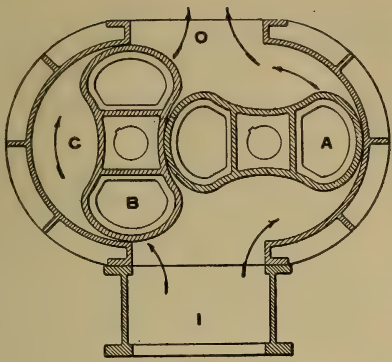


FIG. 420. Two-Lobe Cycloidal Pump.

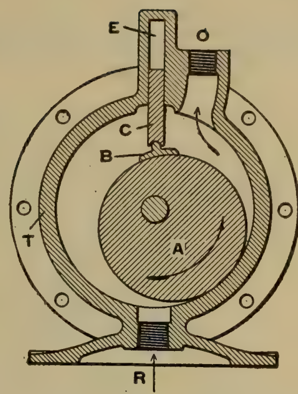
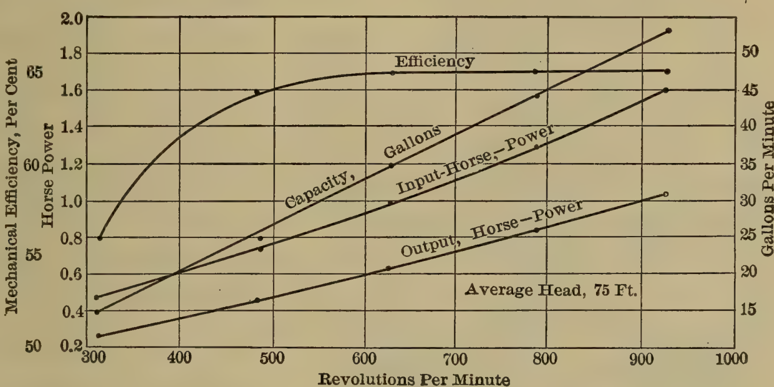
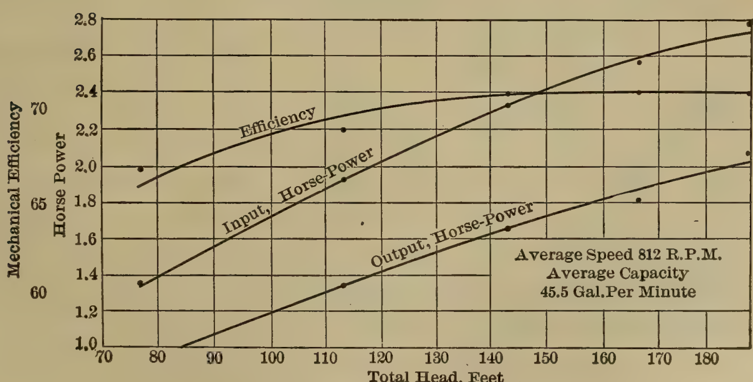


FIG. 421. Rotary Pump with Movable Butment.



Head Constant, Speed Variable.



Speed Constant, Head Variable.

Fig. 422. Performance of a Small Rotary Pump.

give much higher efficiencies, but the general characteristics are about the same. A combined efficiency of pump and engine as high as 84 per cent has been recorded. (Trans. A.S.M.E., Vol. 24, p. 385.)

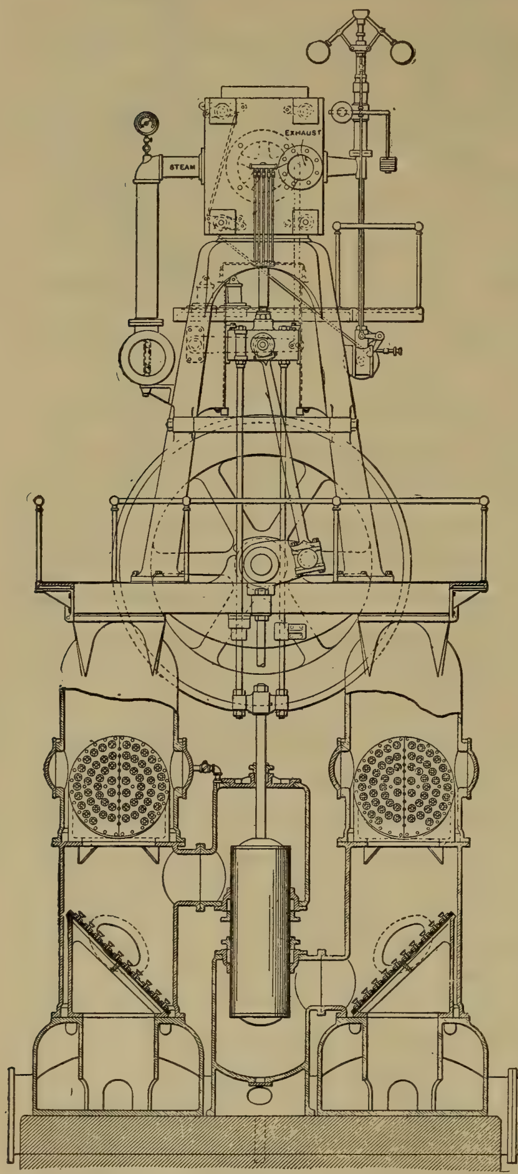


Fig. 423. 10,000,000-Gallon Circulating Pump.

335. Circulating Pumps. — This term is ordinarily applied to the pumps which supply injection water to the condenser.

Three types are found in practice: the piston, the centrifugal, and the rotary pump. Figs. 296 and 309 show the application of reciprocating pumps to condenser installations and Figs. 312 and 313 a similar application of centrifugal pumps.

For large volumes of water and low heads the centrifugal or rotary pump is generally adopted on account of minimum space requirements and low first cost.

In very large central stations where the demand for circulating water is enormous and the lift is moderately high, the high-duty pumping engine is sometimes installed. Fig. 423 shows a section through one of the nine high-duty circulating pumps at the New York Rapid Transit Company's power house. The steam end is operated by Corliss cylinders and is of the cross-compound type. The maximum capacity is 10,000,000 gallons per day

(24 hours) against a head of 50 feet at mean low water. The actual lift is much less than this, as the discharge is aided by the vacuum in the condenser.

336. Hot-well Pumps. — In surface-condenser practice the condensed steam is often handled by a small independently driven pump called the *hot-well pump*. The piston type of pump is being rapidly superseded by the turbine-driven centrifugal pump in this connection. The water from the condenser hot-well should flow to the pump under a head of 3 or 4 feet or more, if convenient. If the head on the suction side is less than this the pump "cavitates" or becomes vapor bound and is unable to remove the water. The discharge over the pump should be provided with a check valve to prevent water returning to the condenser. Centrifugal hot-well pumps are ordinarily designed with horizontal suction and vertical discharge to minimize air pockets in the volute. These pumps are ordinarily operated without automatic control and are permitted to operate at constant speed.

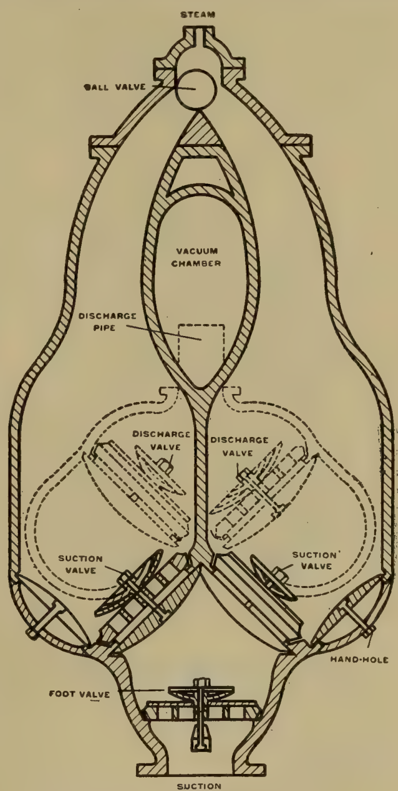


FIG. 424. The Pulsometer.

337. Air Lift. — The air lift is a simple arrangement of piping whereby water may be raised by means of compressed air. There are no working parts, and no valves are employed except to regulate the supply of air. Its particular field of application lies in pumping water from a number of scattered wells, and on account of the total absence of working parts it is peculiarly adapted to handling water containing sand, grit and the like. The device consists of a partially submerged water pipe and air supply variously arranged as in Fig. 425 (A) to (D). Compressed air forced into the water pipe at or near the bottom decreases the density of the column and the difference in weight between the solid column of water B and the air-water column A causes the flow. The successful operation of this device depends upon the ratio of the depth of submersion B to the total head C.

The quantity of air necessary to operate an air lift may be closely approximated from the equation (see Prac. Engr. U. S., April 1, 1912, p. 354)

$$V = \frac{L}{\log \frac{S + 34}{34} \times C}, \tag{250}$$

in which

V = cubic feet of free air per gallon,
S = actual submergence in feet,
C = coefficient determined from experiment.

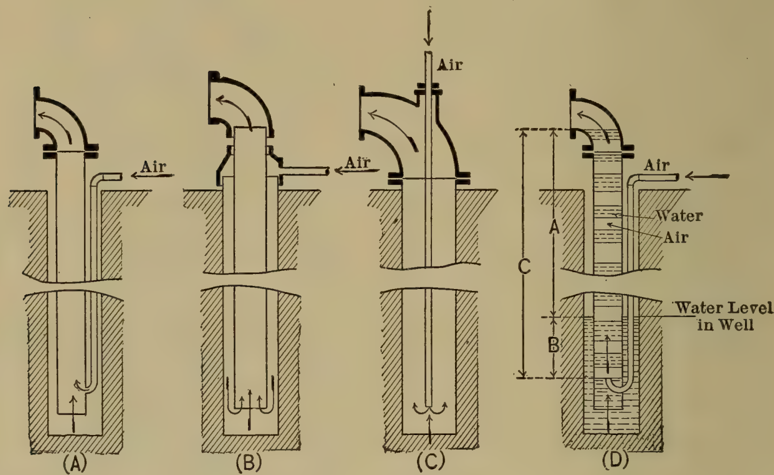


FIG. 425. Different Arrangement of the "Air Lift."

The actual submergence S may be determined from the relationship

$$S = \frac{LS_p}{l_p}, \tag{251}$$

in which

L = actual lift in feet (A, Fig. 425),
S_p = submergence, percentage ($100 \frac{B}{C}$, Fig. 425),
l_p = lift, percentage ($100 \frac{A}{C}$, Fig. 425).

The coefficient C may be approximated as follows:

$$C = 255 - 0.1\,L. \tag{252}$$

For the air pressure required for any lift and any percentage of submergence it is convenient to divide the actual submergence in feet by 2 to get the gauge pressure in pounds. This gives enough pressure in

excess of that due to water head to allow for the pipe friction and other losses.

The efficiency ("water" horse power divided by "air" horse power) varies from 30 to 50 per cent, increasing as the ratio $\frac{B}{C}$ increases from 0.55 to 0.85. (Engineer, U. S., Aug. 15, 1904, p. 564.) A number of tests gives efficiencies ("water" horse power divided by i.h.p. of steam cylinder) varying from 20 to 40 per cent. The horse power required to compress one cubic foot of free air to different pressures per square inch, as determined from actual practice, is approximately as follows:

Pressure in Pounds.	Horse Power Required to Compress 1 Cubic Foot.	Pressure in Pounds.	Horse Power Required to Compress 1 Cubic Foot.
176	0.434	60	0.159
140	0.376	45	0.145
100	0.201	30	0.121
80	0.189		

(Engr., Lond., Aug. 14, 1903, p. 174, Dec. 11, 1903, p. 568, Feb. 12, 1904, p. 172.)

When it becomes necessary to raise water to a height exceeding say 175 feet above the level in the well, it is customary to use two or more pumps, the total lift being divided between them.

Air Lift: Eng. and Contr., Feb. 14, 1912; Bulletin No. 450, Univ. of Wis.; Prac. Engr., April 1, 1912; Power, May 21, 1912.

Pulsometer: Tech. Quar., Sept., 1901; Public Works, Aug. 15, 1904; Engr. U. S., July 15, 1904; Experimental Eng., Carpenter, p. 621; Thermodynamics, Wood, p. 293; Trans. A.S.M.E., 13–211.

Cost of Operating American Pumping Stations: Eng. Rec., Aug. 6, 1904; Proc. Engrs. Club of Phil., Oct., 1906.

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The Selection of Waterworks Pumping Machinery: Eng. and Contr., April 24, 1912.

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The Hydraulic Ram: Engr. and Contr., Jan. 3, 1912; Mar. 22, 1911.

CHAPTER XIV.

SEPARATORS, TRAPS, DRAINS.

338. Live-steam Separators. General.—The function of a steam separator is the removal of entrained water from steam.

Unless a boiler is liberally provided with superheating surface, the steam may contain an amount of moisture varying from 0.3 to 5 per cent. If the boiler is poorly proportioned or forced far above its rating, this percentage may be greatly increased. The quality of the steam is still further reduced by condensation in the steam pipe, which may vary from 1 to 10 per cent, depending upon the length of pipe and efficiency of covering.

One of the effects of moisture in steam is to increase its density and reduce its elastic force. It also increases its conductivity, so that during the work of expansion more heat is absorbed from the walls of the cylinder and discharged into the atmosphere or into the condenser without doing useful work. (Ewing, "The Steam Engine," p. 151.) Although the heat loss from this cause is small, the danger arising from the introduction of a considerable amount of water in the cylinder renders the removal of the moisture necessary. See par. 193 for influence of moisture on steam consumption.

The essentials of a good separator are high efficiency as a water eliminator, ample storage capacity for any sudden influx of water, simplicity and durability in construction, and small resistance to the current of steam passing through. A good separator may be relied upon to remove practically all of the moisture from steam containing under ten per cent entrainment and all but two per cent from steam containing as much as twenty per cent. (Engineer, U. S., Jan. 15, 1904.)

Table 105 gives the results of a series of tests made by Professor R. C. Carpenter in 1891 of six steam separators. (Power, July, 1891, p. 9.) Conclusions from these tests were:

1. That no relation existed between the volume of the several separators and their efficiency.

2. No marked decrease in pressure was shown by any of the separators, the most being 1.7 pounds by separator *E*.

3. Although changed direction, reduced velocity, and perhaps centrifugal force are necessary for good separation, still some means must be provided to lead the water out of the current of the steam.

A series of tests made at Armour Institute of Technology in 1905 on a number of separators showed that the *efficiency of separation decreased as the velocity of the steam increased.** At the low velocity of 500 feet per minute all separators were equally efficient, at a velocity of 5000 feet per minute several had little effect on eliminating the moisture present, and at a velocity of 8000 feet per minute only one gave efficient results.

TABLE 105.

TESTS OF STEAM SEPARATORS.

(R. C. Carpenter.)

Make of Separator.	Test with Steam of about 10 Per Cent of Moisture.			Tests with Varying Moisture.		
	Quality of Steam Before.	Quality of Steam After.	Efficiency.	Quality of Steam Before.	Quality of Steam After.	Average Efficiency
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
B.	87.0	98.8	90.8	66.1–97.5	97.8–99	87.6
A.	90.1	98.0	80.0	51.9–98	97.9–99.1	76.4
D.	89.6	95.8	59.6	72.2–96.1	95.5–98.2	71.7
C.	90.6	93.7	33.0	67.1–96.8	93.7–98.4	63.4
E.	88.4	90.2	15.5	68.6–98.1	79.3–98.5	36.9
F.	88.9	92.1	28.8	70.4–97.7	84.1–97.9	28.4

339. Classification of Separators. — Separators are based on one or more of the following principles of action:

1. *Reverse current.* The direction of the flow is abruptly changed, usually through 180 degrees. This causes the water in the steam, on account of its greater specific gravity, to be thrown into a receiving vessel, while the steam passes on in a reverse direction.

2. *Centrifugal force.* A rotary motion is imparted to the steam whereby entrained water particles are eliminated by centrifugal force.

3. *Baffle plates.* The flow is interrupted by corrugated or fluted plates to the surfaces of which the water particles adhere and from which they fall by gravity to the well below.

4. *Mesh.* The separation is brought about by mechanical filtration through screens or meshes.

The following outline shows the classification of typical separators, in accordance with the above principles:

* See Power, May 11, 1909, p. 834.

Live-steam separators	Reverse current	Hoppes, Fig. 426. Stratton, Fig. 427.
	Centrifugal	Keystone, Fig. 428. Mosher. Robertson.
	Baffle plate	Bundy, Fig. 429. Austin, Fig. 430. Detroit.
	Mesh	Direct, Fig. 431. Potter.
Exhaust-steam separators	Jacketed baffle	Baum, Fig. 432.
	Absorption	Loew, Fig. 433.

340. Reverse-current Steam Separators. — Fig. 426 shows a section through a Hoppes steam separator and illustrates the principle of reverse-current separation. Steam may flow through in either direction. Both the inlet and outlet ports are surrounded by gutters *C*, *C*, partly filled with water, which intercept the moisture following the surface of the pipe, while the downward plunge of the steam throws the entrained water to the bottom of the separator. The condensation is carried from the troughs by pipe *P* to the well below, from which it is trapped at *D* in the usual way. The velocity of the steam in passing through this separator is greatly reduced to prevent the steam from taking up the water in the bottom of the well. This is brought about by increasing the area of the passage through the separator.

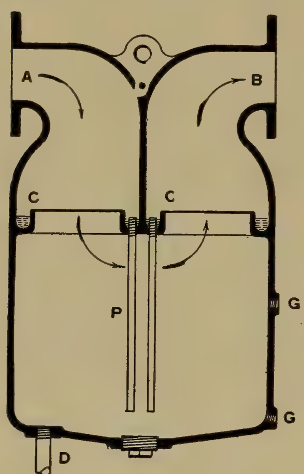

FIG. 426. Hoppes Steam Separator.

Fig. 427 gives a sectional view of a Stratton separator, which, though primarily of the reverse-current type, embodies also the principle of centrifugal force. The separator consists of a vertical cast-iron cylinder with an internal central pipe *C* extending from the top downward for about half the height of the apparatus, leaving an annular space between the two. The current of steam on entering is deflected by a curved partition and thrown tangentially to the annular space at the side, near the top of the apparatus. It is thus whirled around with all the velocity of influx, producing the centrifugal action which throws the particles of water against the outer cylinder. These adhere to the surface, so that the water runs down continuously in a thin sheet around the outer shell into the receptacle below. The steam, following in a spiral course to the bottom of the internal pipe, abruptly enters it, and

passes upward and out of the separator without having once crossed the stream of separated water. The rapid rotation of the current of steam imparts a whirling motion to the separated water which tends to interfere with its proper discharge from the apparatus. The separator has therefore been provided with wings or ribs *E* projecting at an acute angle to the course of the current, which have the effect of breaking up this whirling motion and allowing the water to settle quietly at the bottom, whence it passes off through the drain pipe *D*.

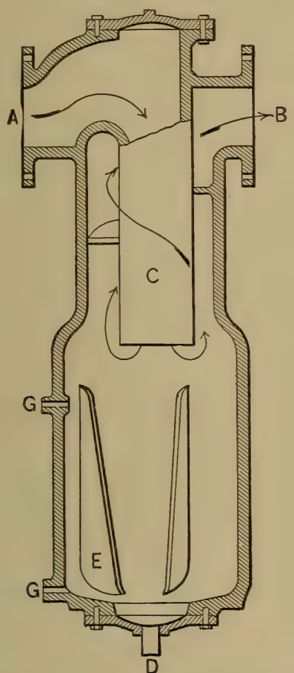

FIG. 427. Stratton Steam Separator.

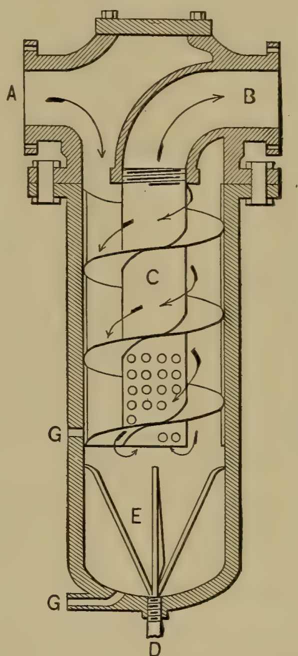

FIG. 428. Keystone Steam Separator.

341. Centrifugal Steam Separators. — Fig. 428 shows a section through a Keystone or Simpson's centrifugal separator. The separator consists of a cast-iron cylinder with vertical pipe *C* extending downward about two-thirds of the whole length; this pipe has a thread or screw wound spirally around it, the space between the threads being somewhat greater than the area of the steam pipe. The steam passing around the spiral course causes the water to be thrown against the outer walls by centrifugal force, while the dry steam passes through the small holes in the central pipe. The water passes down the outer walls, where its motion is arrested by obstructing ribs *E*, and is thence carried away by a drip pipe *D* to a suitable drain.

342. Baffle-plate Steam Separators. — Fig. 429 gives an interior view of a Bundy separator and illustrates the application of baffle plates for live-steam separation. This separator consists of a rectangular cast-iron casing with a cylindrical receiver beneath it. Directly across the steam passage are baffle plates corrugated for the reception of entrained water. The plates consist of vertical castings, each containing a main artery or channel which leads directly to the receiver. The fronts of the plates are flat, with a series of recesses sloping inwards and downwards, terminating in an opening of capillary size leading to the main artery. The plates are staggered, so that the steam must impinge against all of them in its passage. The particles of water adhere to the plates, collect, and fall by gravity into the receiver. The flanges at the bottom constrict the opening of the reservoir so as to prevent the steam from picking up any portion of the water.

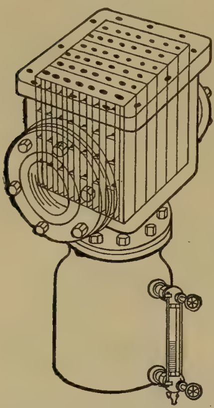

FIG. 429. Bundy Steam Separator.

Fig. 430 shows a section through an Austin separator and illustrates another class embodying the fluted baffle-plate principle. The steam in passing through the chamber impinges against the fluted baffle plate *B*. The moisture adheres to the surfaces, collects and trickles along the corrugations to the bottom of the well. These corrugations are formed in such a manner that the steam cannot come in contact with the water particles after they have been once eliminated. A perforated diaphragm *D* prevents the water in the well from coming in contact with the steam. The current of steam is also reversed, thus giving additional separating properties to the apparatus.

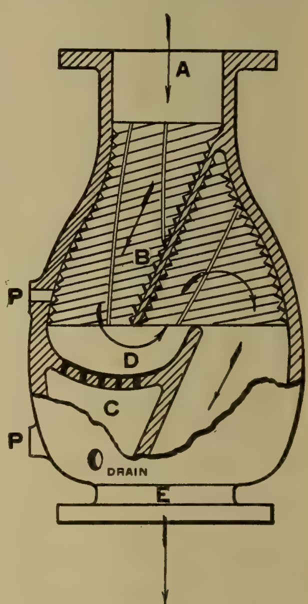

FIG. 430. Austin Steam Separator.

343. Mesh Separators. — Fig. 431 shows a section through a "direct" separator, illustrating the principle of mesh separation. These separators are made with steel bodies and cast-iron heads and bases, in all sizes up to six inches inclusive, the larger sizes being constructed of cast iron or boiler plate. The cone *C*, perforated lining *E*, and diaphragm *S* are made of cold-rolled copper; the cone *O*

is a substantial gray-iron casting, resting on three cast-iron supports hooked over the top of inner pipe as indicated. The method of operation is as follows: The accumulated moisture around the walls of the steam pipe is caught by the upper edge of cone *C* and carried down back of lining *E* to the water chamber. The current of steam entering the separator impinges upon the conical surface, which is composed of solid plate *O* covered with sieve *S*, through which water may freely pass but from which it cannot readily escape. Passing through the sieve and depositing on the solid surface of the cone *O*, this water is carried by conductors *P* to the water chamber. Perforated lining *E* permits the moisture content of the steam to pass through the opening to the water below and prevents it from coming in contact again with the current of steam. A trough is provided at the lower edge of the inverted cup which leads all the water that may adhere to it to the water chamber. The steam flows through the passages indicated by arrows and is subjected to a whipsnapping action which tends to throw off any remaining moisture. The perforated plate *D* prevents the steam from picking water out of the water chamber.

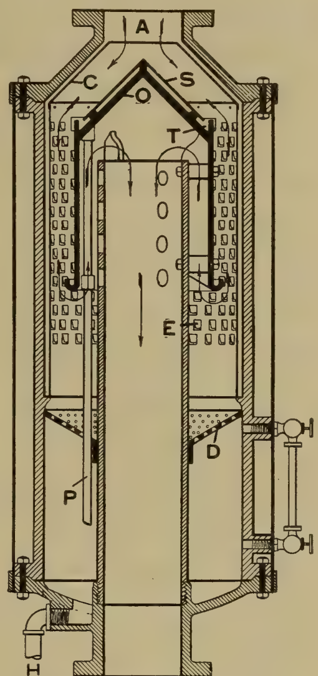

FIG. 431. "Direct" Steam Separator.

344. Location. — Live-steam separators may be located

1. Inside the boiler,
2. Between boiler and engine,
3. At the steam chest.

Where the steam pipe is very short, and particularly in marine and locomotive work where the tossing of the boiler induces excessive priming, the separator may be placed inside the boiler and its function becomes that of a dry pipe. In this location it prevents the water due to foaming and priming from passing to the engine, and reduces condensation in the pipe by supplying dry steam. The "Potter mesh" and the "De Rycke centrifugal" are types of separators designed for this service.

The arrangement of separator between engine and boiler, other than at the throttle or inside the boiler, is sometimes necessary for economy

of space. Where possible, however, the separator should be placed close to the steam chest.

Current practice recommends that a receiver separator, which is an ordinary separator with a volume of two to four times that of the high-pressure cylinder, be placed close to the engine if the load is intermittent or sharply fluctuating. This forms a cushion for absorbing the force of the blows caused by cut-off, delivers steam at a practically uniform pressure, and reduces the vibration of the piping to a minimum. It also provides a reservoir for sudden demands made by the engine. Smaller pipes and higher velocities may be used with this arrangement.

345. Exhaust-steam Separators and Oil Eliminators. — The function of an exhaust-steam separator is the removal of cylinder oil from the steam exhausted by engines and pumps. In plants where exhaust steam is used for heating it is quite essential to remove the oil from the steam before it enters the heating system, for the oil not only reduces the efficiency of the radiators by coating them with an excellent non-conducting film but is an element of danger to the boiler itself. In condensing plants the separator will prevent the oil from fouling the condenser tubes and those of the vacuum heater if one is installed; this is an important factor, since the oil or grease lowers the efficiency of the heat transmission.

In a general sense a live-steam separator is also an oil eliminator, and all the separators previously described perform this function to a certain extent, since the underlying principles governing the elimination of oil from exhaust steam are similar to those employed in removing water from steam. Most of the separators described above are also designed in lighter form, as oil eliminators, but by far the greater number are based on the fluted baffle-plate principle, of which the Hine, Bundy, Cochrane, Utility, Peerless, and Keiley are well-known examples. This type of oil separator will eliminate a considerable portion of the oil in the steam, provided the baffle plates or corrugated surfaces are frequently cleaned.

The following is taken from the report of Professor R. Burnham of the Armour Institute of Technology on the test of a six-inch horizontal oil separator of the baffle-plate type:

"For purposes of test the separator was placed in the exhaust line of a 9 × 18 × 24 cross-compound Corliss engine running under its maximum load at 80 pounds pressure and exhausting into a Wheeler surface condenser against 26 inches vacuum.

"Cylinder oil was fed through the lubricators of the high and low pressure cylinder at the rate of from 5 to 20 drops per minute, a record

being made of the exact quantity of oil fed per hour. The separator was so arranged, by means of a receiver connected to the air pump, that the accumulation of oil and water could be readily trapped from it at any time. In order to determine the quantity of oil given up by the condenser, and not properly charged against the separator, each series of efficiency tests was preceded by a run of three hours during which time no oil whatever was fed to the cylinders. During the last hour a record was made of the weight of steam used and a sample of the condenser discharge retained for analysis.

"The efficiency tests were made by feeding at an excessive rate through the lubricators as described above, and when conditions became practically constant, records were made for one hour of the weight of oil used, weight of condensed steam, and drain from separator. Samples of the two latter were retained for analysis and the percentage of oil in them accurately determined, correction being made for the oil given up by the condenser. A second series of tests was made exhausting at atmospheric pressure. The results obtained are tabulated below.

	Exhausting into 26-inch Vacuum.			Exhausting at Atmospheric Pressure.	
Oil in condensed steam with no oil feeding. (Charged to condenser.) Pounds per hour	.051	.057	.0559	.0353	.0340
Oil fed to cylinder, pounds per hour.......	.401	.562	.934	.621	.710
Steam condensed per hour, pounds	1000	1120	1096	905	872
Oil caught by separator, per hour, pounds A	.341	.450	.743	.552	.583
Oil in condensed steam (corrected), pounds per hour............................B	.009	.010	.0096	.0071	.0050
Percentage of oil in condensed steam by weight, per cent........................	.0009	.001	.00088	.00078	.00057
Efficiency of separator, per cent $\frac{A}{A+B}$ ...	97.4	97.8	98.8	98.7	99.1

"There was practically no free oil on the surface of the condenser discharge in any case, the small quantity of oil which passed the separator (from 5 to 10 parts in a million of water by weight) existing as an emulsion, imparting a slight milky color to the water."

It is a well-established fact that oil can be more effectually removed from wet than from dry steam, and some makers, notably the Austin Separator Company, inject a cold-water spray into the separator chamber. A similar result is brought about in the Baum separator, Fig. 432, in which the corrugated baffle plate is hollow and cold water is forced through the chamber thus formed. Referring to Fig. 432: The diverged baffle plate forms the wall of a chamber in which cold water is con-

tinually circulated. This circulation causes moisture to appear on the baffle-plate surface. The particles of oil, coming in contact with this moist surface as the steam current is diverged, adhere to it and fall by gravity into the well below, where they are completely isolated from the purified steam. A large portion of the oil and water, however, does not enter the separator at all but is caught by the inside ledge near the junction of the exhaust pipe and the separator. The oil and condensation which are carried along the bottom of the pipe come in contact with this ledge and are carried directly to the outlet pipe.

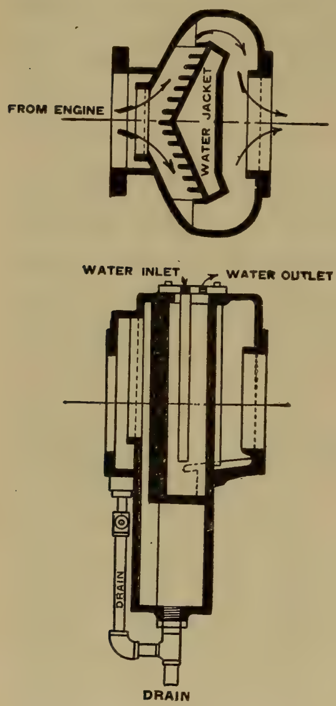

FIG. 432. Baum Oil Separator.

A very successful method of removing oil from steam is to project the steam on to the surface of a body of water. The water may be hot or cold and will hold the oil if it once reaches the surface. It is essential, however, to reduce the velocity of the steam as it passes on its way to the outlet. Baldwin's grease separator is based upon this principle. (Baldwin on Heating, p. 234.)

The most efficient method of removing oil is by combined filtration and absorption. (Engineering News, May 22, 1902, p. 406.) A large chamber filled with coke, brick, broken tile, or other absorption material is placed in series with the exhaust pipe. The steam passing through this chamber is entirely freed from oil and moisture, provided the absorbing material is sufficient in quantity and is replenished as soon as it becomes saturated with oil. The annoyance attending the removal and replenishing of the absorbing material at frequent intervals and the great size of the apparatus are serious drawbacks. An example of this system of purification in which many of the objectionable features are reduced to a minimum is the Loew grease and oil extractor, Fig. 433. The exhaust steam enters the chamber at the top, strikes a large deflecting plate shaped like an inverted V, and permits part of the condensation and oil to be drawn off by the drain pipe. The steam then rises and is deflected, as indicated, against a series of shelves filled with fibrous material covered with coarse wire screens. The grease is removed from each shelf by suitable drains. This apparatus is

sectional and any number of sections may be added without affecting the rest.

In a non-condensing plant where the exhaust steam is used for heating purposes the oil separator is ordinarily placed in the main exhaust pipe just before it enters the heating system. Where several branches enter one main it is not customary to place a separator in each branch, one large separator located as above being sufficient.

In condensing plants oil separators are seldom installed except where surface condensers are used, in which case the separator may be placed anywhere between the engine and condenser. In case a vacuum heater is used the separator may be placed on either side of the heater, depending upon the type of separator. If the separator is of the "jacket-cooling" or "spray" type, it may be placed between the engine and the vacuum heater; if, however, it is of the "baffle-plate" type, the oil will be more efficiently removed if the separator is placed between the heater and condenser so that it will get the benefit of the moisture formed in the heater. In the latter location, however, the separator will not prevent the oil from fouling the heater tubes.

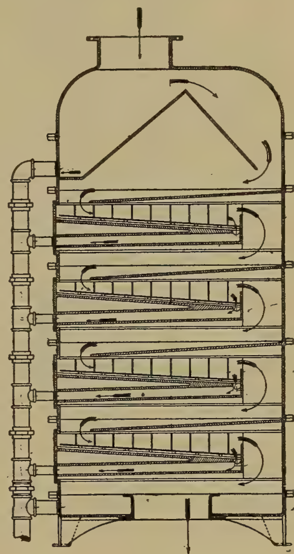

FIG. 433. Loew Grease Extractor.

Where a jet condenser is used and water is taken from the hot well, the hot well itself acts as an oil separator. (Trans. A.S.M.E., 24–1144.)

All separators, steam and oil, should be provided with gauge glasses and should be thoroughly drained and the drainage should be automatic.

346. Exhaust Heads. — The function of the exhaust head is the elimination of oil and water from steam exhaust before permitting it to be discharged into the atmosphere. Unless removed, the water and oil rot the roofs and walls in summer and pollute the atmosphere surrounding the plant. The exhaust head also acts as a muffler reducing the noise of the escaping steam. Exhaust heads are built on the same principle as steam and oil separators and most separator builders manufacture them. Fig. 434 shows a section through a typical exhaust head. The condensation is ordinarily drained to waste, though with proper purification it may be returned to the boiler. With an efficient oil sepa-

rator in the exhaust line the condensation in the exhaust head may be returned directly to the boiler without further purification.

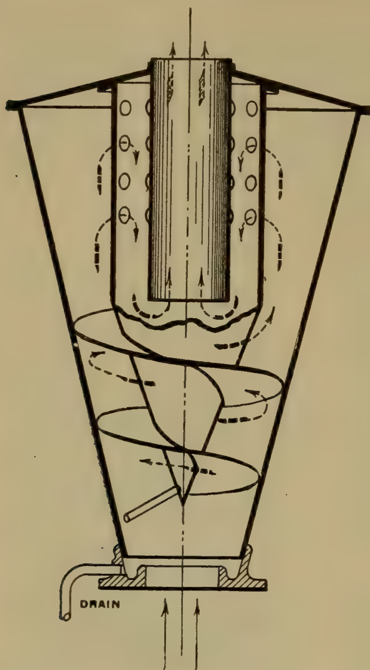

Fig. 434. A Typical Exhaust Head.

Live-steam separators are proportioned so that it is only necessary, in the average installation, to specify the size of pipe, the type of engine, the steam pressure, and the style, whether horizontal or vertical. Gauge glasses, gauge cocks, and companion flanges are usually provided by the maker. In some cases the capacity of the reservoir is also specified. In specifying oil extractors the following additional data are necessary for an intelligent choice: the number of engines and pumps exhausting into the line, the location of the separator, the steam pressure, *velocity*, and the quality and quantity of cylinder oil used. A guarantee of efficiency and of material and workmanship is often demanded.

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The Hot Well as an Oil Extractor: Trans. A.S.M.E., 24–1144.

347. Drips. — No matter how thoroughly a steam pipe or reservoir may be covered with insulating material considerable condensation takes place. With the best covering this loss approximates one sixth of a pound of steam per square foot of pipe surface per hour for steam pressures of one hundred pounds, and runs as high as one pound of steam for bare pipes. See Table 113 for results of experiments on the loss of heat from bare pipes, and Table 114 for data on the efficiency of pipe coverings. In addition to this water of condensation, from $\frac{1}{2}$ to 2 per cent of moisture is carried over by the steam from the boiler. This water, unless thoroughly removed, is a constant source of danger to the engines and causes water hammer and leaky joints in the piping.

A joint on a steam pipe may safely withstand a steam pressure of 100 pounds without leaking and still leak badly under a water pressure of half that amount. This is due to the fact that the steam with its high temperature causes the pipe to expand, thus insuring a tight joint, while the entrained water (which cools as it collects) causes the pipe to contract and allows a leak.

The entrained water and water of condensation are usually spoken of as "drips." Drips may be divided into two classes, low pressure and high pressure.

348. Low-pressure Drips. — Low-pressure drips include the steam condensed in heating systems, exhaust steam feed heaters of the close type, exhaust steam piping, receiver barrels, steam chests, and exhaust heads. As these drips are impregnated with oil and are useless for boiler feed without purification, they are usually discharged to waste. Most city ordinances require the drips to be cooled to 100 degrees F. before being discharged into the sewer. In this case they must be first discharged into a tank and permitted to cool. This tank must be vented to the atmosphere to prevent back pressure. Fig. 435 shows an installation in which the heat abstracted from the drips, etc., is used to heat the feed water. The drips from the throttle valve and steam chest in a non-condensing plant are ordinarily discharged into the exhaust pipe as shown in Fig. 436. In a condensing plant the throttle drips are piped to a trap or to the free exhaust pipe. The re-

turns from a steam-heating system are sometimes classified as low-pressure drips. They are invariably returned to the boiler.

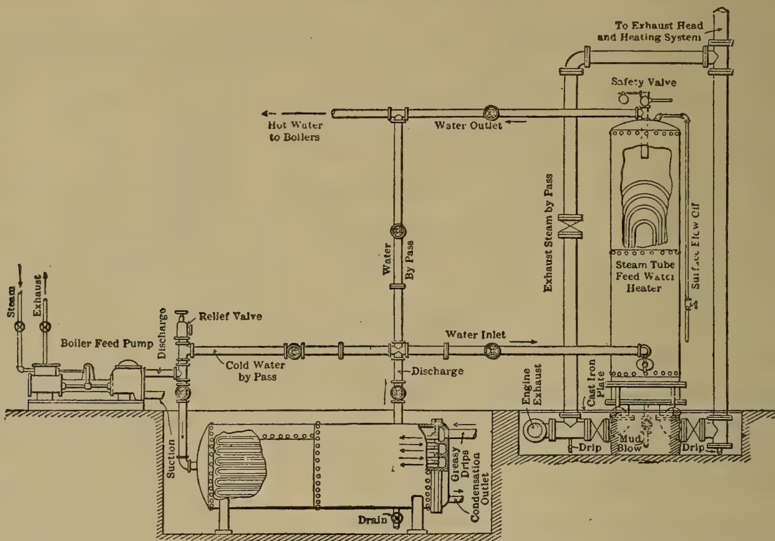


FIG. 435. Closed Heater Installation for Abstracting Heat from Oily Drips.

In small plants all the low-pressure drips may be connected to one large pipe and this pipe in turn to a single trap, provided there is but little difference in pressure in the various drip pipes. In case of different pressures separate leads should be run to waste or traps.

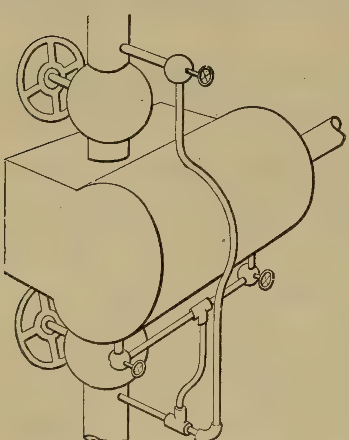


FIG. 436. Simple Method of Draining Drips.

The drips from the receiver and vacuum heater barrels in a condensing plant are oftentimes under less than atmospheric pressure, and sometimes the pressure varies from a slight vacuum to 10 or 20 pounds gauge, and consequently cannot be disposed of as described above. If possible, the heaters and receivers should be placed so as to drain into the condenser (see Fig. 449). Should this arrangement prove impracticable, the barrels may be drained by a trap especially arranged as shown in Fig. 450.

349. Size of Pipe for Low-pressure Drips. — In the average exhaust-steam feed-water heater one pound of steam in condensing gives up

approximately 1000 heat units. This will heat about 6 pounds of water from 60 to 200 degrees F. Hence the area of the drip which carries the water of condensation from the closed heater need be but one fifth that of the feed pipe. In no case, however, should a pipe smaller than one half inch in diameter be used. Should the same pipe be used for both exhaust head and heater drips, an area of one fourth area of feed pipe would prove of ample capacity. In practice it is customary to use the size of pipe conforming with the outlet furnished by the manufacturer of the apparatus, and only when several pieces of apparatus are connected to one main are calculations made for the size of this main.

The drip pipe from the throttle valve is ordinarily one half inch in diameter irrespective of the size of steam pipe; this is also true of the steam-chest drip.

350. High-pressure Drips. — High-pressure drips consist of those which are condensed under practically boiler pressure and include the steam condensed in steam pipes, cylinder jackets of engines, reheating coils of receivers, and separators. Being free from oil and containing considerable heat, they are usually returned to the boiler. Drips may be returned to the boiler automatically by means of

1. Steam traps.
2. Holly steam loop.
3. Pumps.

351. Classification of Steam Traps. — Steam traps may be divided into two classes, depending on their use, — return and non-return. Both of these two classes may be subdivided into five types according to the principle of operation, viz.:

I. Float. III. Bowl.
II. Bucket. IV. Expansion.
V. Differential.

Return Traps.

Traps which receive the condensed steam and return it to a boiler having considerably higher pressure than that acting on the returns are known as *return traps*. They are made in a great variety of styles. The general principle of operation is shown in Fig. 446 and described in paragraph 356.

Non-return Traps.

Non-return traps, as the name implies, are used where the water of condensation is not returned to the boiler but is discharged into any receptacle having less than boiler pressure.

CLASSIFICATION OF A FEW WELL-KNOWN STEAM TRAPS.

- Steam Traps
  - Float
    - McDaniel.
    - Cookson.
  - Bucket
    - Acme.
    - Albany.
  - Dump
    - Bundy.
    - Morehead.
  - Expansion
    - Metal
      - Columbia.
      - Geipel.
    - Volatile-Fluid
      - Dunham.
      - Heintz.
  - Differential
    - Flinn.
    - Siphon.

352. Float Traps. — Fig. 437 shows a section through a McDaniel improved trap, illustrating the principles of the float type. A hollow

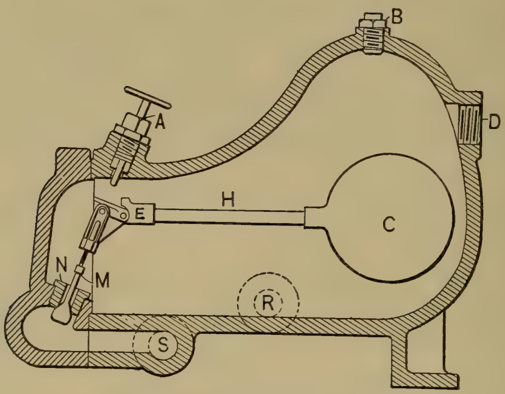


FIG. 437. McDaniel Float Trap.

sphere *C* of seamless copper pivoted at *E* rises and falls with the change of water level in the vessel. The discharge valve *M* is operated by the float. When the trap is empty the float is in its lowest position and the discharge valve is closed. Water of condensation flows into the trap by gravity through opening *D* to a certain depth, when the float opens the discharge valve and the steam pressure acting on the surface of the water forces it through outlet *S* to tank or atmosphere. After the water is discharged the float closes the valve and permits the condensation to collect again. A gauge glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is a danger of considerable steam leakage through the discharge valve, due to unequal expansion of valve and seat and the sticking of moving parts.

The discharge from a float trap is usually continuous, since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present. When the trap is working lightly, this adjustment is apt to throttle the area and create such a high velocity of discharge as to cause a rapid wear of valve and seat. This defect is more or less evident in all steam traps discharging continuously. For this reason all wearing parts should be accessible and readily replaceable.

353. Bucket Traps. — Fig. 438 shows a section through an "Improved Acme" steam trap. The water of condensation enters the cast-iron vessel at *A*, filling the space *D* between the bucket *E* and the walls

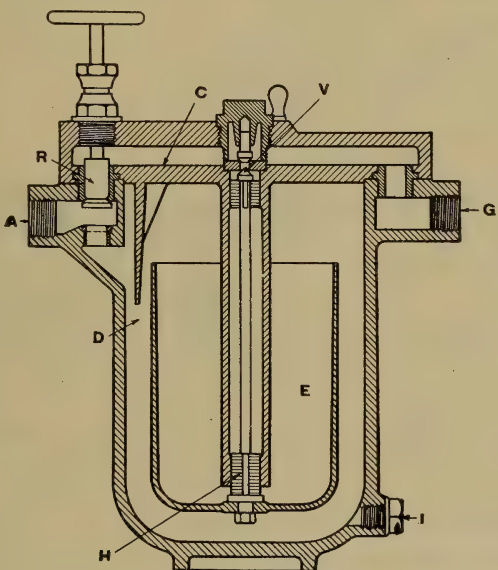


FIG. 438. Acme Bucket Trap.

of the trap. This causes the bucket to float and forces valve *V* against its seat (valve *V* and its stem being fastened to the bucket as indicated). When the water rises above the edges of the bucket it flows into it and causes it to sink, thereby withdrawing valve *V* from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water through the annular space *H* to discharge opening *G*. When the bucket is emptied it rises and closes valve *V* and another cycle begins. By closing valve *R* the trap is by-passed and the condensation blows directly through passage *C* to discharge *G*. The discharge from this type of trap is intermittent.

354. Dump or Bowl Traps. — Fig. 439 shows sections through a Bundy bowl trap of the "return" type. The water enters the bowl

through trunnion D and rises until its weight overbalances counterweight E and the bowl sinks to the bottom. As the bowl sinks, arm G, which is a part of the bowl, rises and engages the nuts N on valve stem H and opens valve I, thus admitting live steam pressure on to the surface of the water. The trap then discharges like all others. After the water is discharged weight E sinks and raises bowl A, which in turn closes valve I, and the cycle begins again. Bowl traps are necessarily intermittent in their discharge.

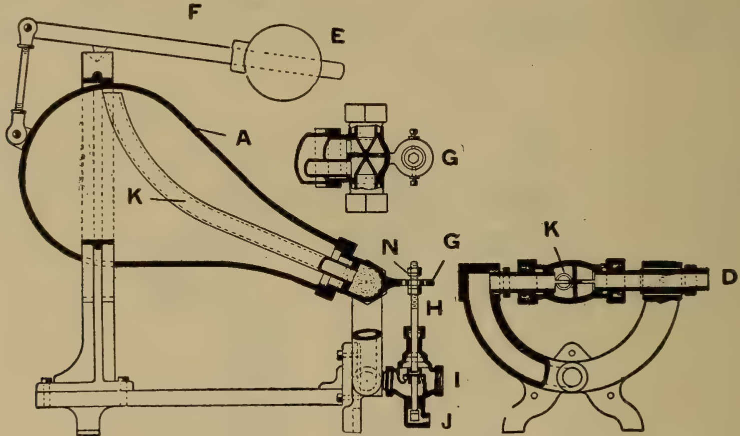


Fig. 439. A Typical Tilting Trap.

Fig. 450 shows the application of a bowl trap to a receiver where the drips are under a vacuum, and Fig. 451 a similar application to an engine receiver where the pressure varies from less than atmospheric pressure to a pressure of 40 or 50 pounds.

355. Expansion Traps. — Expansion traps may be divided into two groups:

(1) Those in which the discharge valve is operated by the relative expansion of metals and

(2) Those in which the action of a volatile fluid is utilized.

Expansion traps will never freeze, as they are open when cold and all the water drains out before the freezing temperature is reached.

Since traps of this type have little capacity for holding water, 5 to 10 feet of pipe should be provided between the trap and the pipe to be drained in order that the condensation may collect and cool.

Fig. 440 shows the general appearance of a Columbia expansion trap in which the valve is operated by the expansion of metallic tubes. Water gravitates to the trap through opening marked "inlet," passes through *brass* pipe O, then downward to the main body of the valves and back to outlet valve C. Below pipe O and parallel to it is an *iron*

rod S, at the end of which is the support or fulcrum of lever R. The lower end of this lever is connected to the stem of the valve C, so that any movement of the lever is communicated to it. When the trap is cold, valve C is open and all water of condensation passes out. The moment steam enters the pipe O it expands. The amount of expansion is multiplied several times by the action of the lever R, so that the

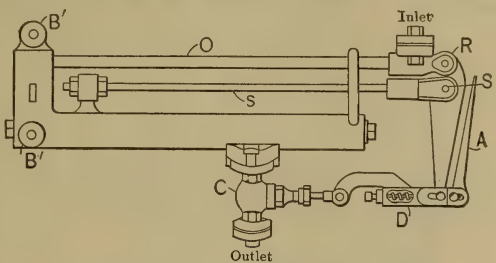


Fig. 440. A Typical Expansion Trap.

movement of the valve is much greater than the expansion of the pipe O. The compensating spring D prevents the brass tube from damaging itself by excessive expansion. Lever A permits the trap to be blown through by hand.

Fig. 441 shows a section through a Geipel trap in which the valve is operated directly by the expansion of two metallic tubes and the

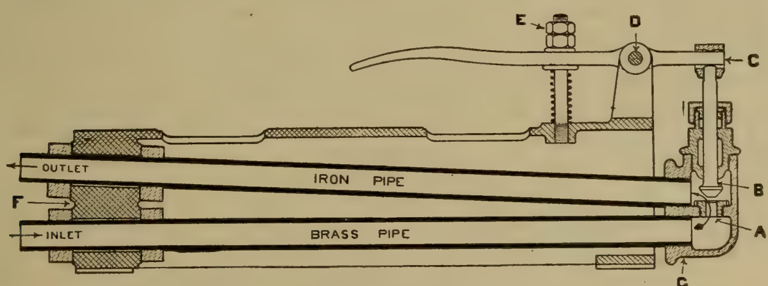


Fig. 441. Geipel Expansion Trap.

movement is not multiplied by levers as with the Columbia. The lower or brass pipe constitutes the inlet and is connected to the vessel to be drained; the upper or iron pipe is the outlet for discharge. The two pipes form the sides of an isosceles triangle, the base F of which is rigid, while the apex A is free to move in a direction at right angles to the linear expansion of the tubes. When cold, the brass pipe is contracted and the apex, in which the valve seat is placed, is moved down so that the valve is open and the water is discharged. As soon as steam enters the brass pipe the latter expands and forces the valve seat against

the valve. The trap may be adjusted for any pressure by means of the lock nuts *E*. When it is desired to blow through, the valve may be operated by hand by pressing the lever.

Fig. 442 shows a section through a Dunham trap. It operates upon the expansion principle, utilizing a fluid of a volatile character as its motive force. The corrugated bronze disk *B* is filled with a volatile fluid, and expands and contracts according to the pressure exerted by

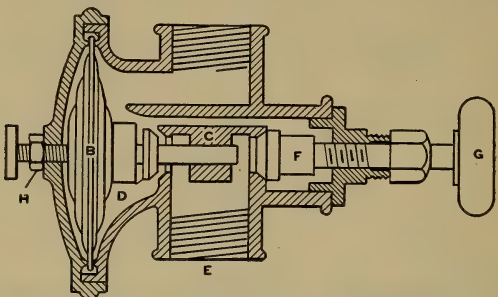


FIG. 442. Dunham Expansion Trap.

the fluid. The water enters at the top, surrounds disk *B* and passes through valve opening *D* to discharge outlet at *E*. As soon as steam strikes the disk *B* the volatile fluid flashes into a vapor and causes the disk to expand. This expansion forces valve *D* against its seat and the discharge ceases. The valve will remain closed until the condensation collects and cools the disk *B*, which then contracts, opens the valve, and condensation enters as before. The adjustment, however, is such that the discharge may be made continuous instead of intermittent.

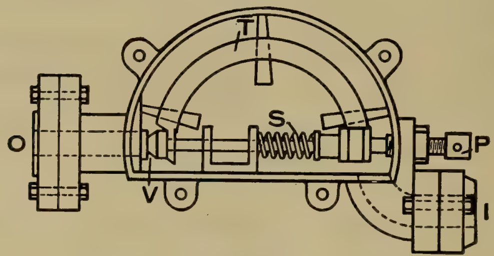


FIG. 443. Heintz Expansion Trap.

The Dunham trap is claimed to be the smallest trap of its capacity on the market. The 1-inch size, having a capacity for draining 10,000 lineal feet of 1-inch pipe under 60 pounds pressure, weighs but 5 pounds and may be connected to the pipe line as if it were a globe valve.

Fig. 443 shows an internal view of a Heintz steam trap. This works on the principle of the volatile-fluid expansion trap but in a different

manner from any of those described above. The requisite movement is obtained by the elongation and contraction of the extremities of a bent metallic tube *T* filled with a highly volatile fluid. This tube is inclosed in a cast-iron box and presses against the point of regulating screw *P*. The other extremity of the tube carries the valve and is free to move under the action of the variations of temperature. Spring *S* has no connection with the action of the trap. It is used as a simple means of holding one end of the expansion tube on its pivot. The trap operates as follows: Water enters at *I*, surrounds the tube *T* and passes through the valve to the discharge outlet *O*. As soon as steam enters the chamber the volatile fluid in the tube flashes into a vapor and the pressure thus created tends to straighten out the tube; this forces the valve against its seat and the discharge ceases. As the trap cools the tube returns to its normal position and the discharge valve is opened, thus permitting the condensation to drain out. The adjustment permits of continuous or intermittent discharge and of variable pressures.

356. Differential Traps. — Fig. 444 shows a cross section through a Flinn differential trap. The column of water *X* acting on diaphragm *D* closes valve *V*. The water entering pipe *E* and the action of the spring equalize column *X* and open the valve. Describing the action in further detail, the water of condensation enters at *A*, fills lower chamber *Y*, pipe *X*, and receiving chamber *C* up to the level of the top of pipe *E*. This column of water acting on the under side of the diaphragm *D* forces the valve to its seat against the counter pressure of the spring *S*. Any additional water that enters the trap overflows through pipe *E*, filling chamber *F* and pipe *E* to a point about midway of its height, where the effect of the column of water in pipe *X* is balanced. The pressure on each side of the diaphragm is then equal, the short column in pipe *E*, aided by the spring, balancing the pressure of the longer column in pipe *X*. Any further increase in the height of the water in pipe *E* causes a depression of the valve *V*, which allows water to escape

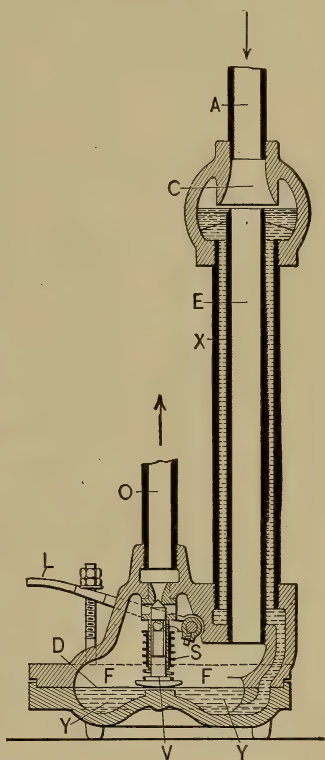


FIG. 444. Flinn Differential Trap.

until the column has fallen to a level a little below the middle of pipe *E*, when this valve closes again. This action is repeated at intervals according to the quantity of water entering the trap. So long as the water keeps coming in sufficiently large quantities the valve remains wide open.

Fig. 445 gives a general view of a siphon trap which is much used in draining low-pressure systems, as, for example, the separator in an exhaust steam heating system. It consists essentially of two legs *A* and *B*, which may be close together or any distance apart but the lengths of which must be sufficiently great to prevent pressure acting through pipe *I* from forcing the water out of *B*. *C* is a vent pipe extending to the air to prevent siphoning; *O* is the discharge for the condensed steam. In ordinary operation the leg *B* is filled with water which is constantly overflowing, and *A* with steam and water, the total pressure in both legs being equal. The siphon trap is applicable for low pressure only, as it requires approximately 2.3 feet of vertical space *E* for each pound per square inch pressure in the pipe. The maximum allowable head is represented by vertical distance *N*.

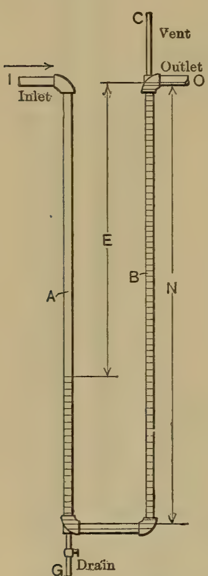


FIG. 445. Simple Siphon Trap.

357. Location of Traps. — Wherever possible a trap should be located so that the condensation will flow into it by gravity. This will insure positive drainage. Sometimes, however, the coils, cylinders, or pipes to be drained are located in a pit or trench or lie on a basement floor where it is impossible to set the trap so as to receive the drains by gravity without placing it in an inaccessible position. With very low pressures this is often unavoidable, but with pressures of five pounds or more the trap may be placed above the point to be drained. If a trap is set in an exposed place a drain should be provided at the lowest point to free the pipe of water when steam is shut off. A dirt catcher or strainer should be placed in the pipe leading to the trap to prevent scale, etc., from reaching the valve. All pockets and dead ends should be drained, and no condensation should be allowed to accumulate. High- and low-pressure drips should be kept separate. All tanks should have gauge glasses.

Fig. 446 shows the application of a float trap for automatically returning water to the boiler. For this purpose the trap must be placed three feet or more above the water line in the boiler, so that the water may gravitate to it. Water is forced into the trap from the returns through

pipe A until it reaches a level where the float opens the equalizing valve V and permits steam from the boiler to enter the trap, thus equalizing the pressures. The water then flows into the boiler by gravity through check valve D. At the end of discharge the float closes the equalizing

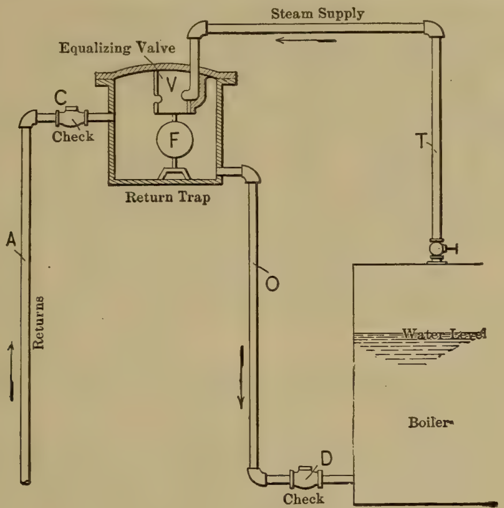


FIG. 446. Return Trap.

valve and another cycle begins. Check valve C prevents the water from being forced back to the return pipe. If the pressure in the return pipe A is not sufficient to force the water into the trap, a pump or another trap may be used to effect this result. Practically any high-pressure trap may be converted into a return trap by proper installation and an "equalizing" valve.

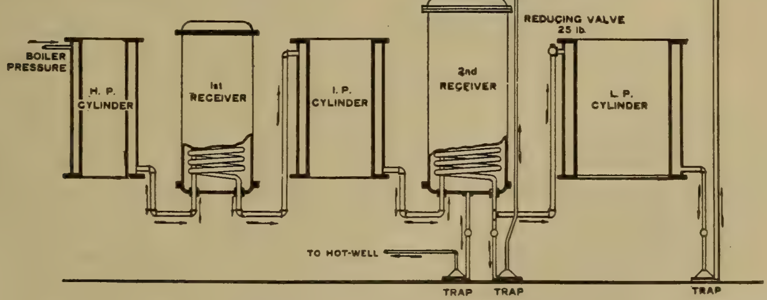


FIG. 447. Drainage System for Jackets and Receivers of Triple-expansion Pumping Engines.

Figs. 447 and 448 show different applications of steam traps to the receiver coils and jackets of triple-expansion pumping engines. The drawings are self-explanatory.

358. Drips under Vacuum. — Conditions frequently make it necessary to remove condensation from apparatus working under a vacuum, as, for example, a primary heater.

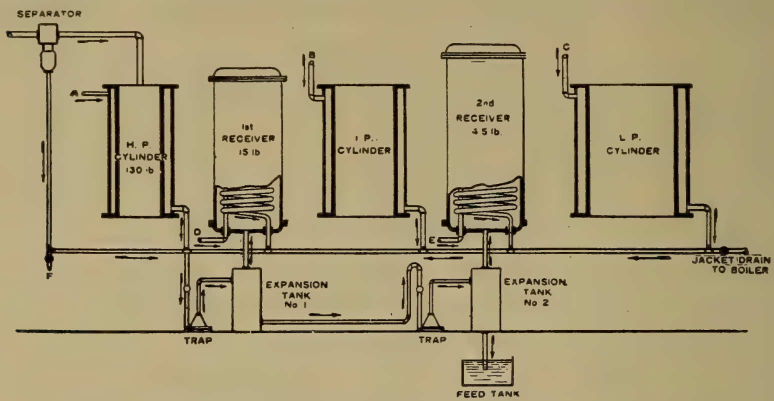


FIG. 448. Drainage System for Jackets and Receivers of Triple-expansion Pumping Engines.

The simplest method is to pipe the drips to the condenser and permit the condensation to gravitate to it as in Fig. 449. Where this is impracticable, as in an installation with the condenser above the heater, a steam trap is usually employed. Fig. 450 shows the application of a Bundy trap to a vacuum or primary heater. A close-fitting weighted check valve W, set to open outwards, prevents intake of air through the discharge pipe while the trap is filling. Connection E is made from the vent underneath the valve stem V back to the heater so as to equalize the pressures. The operation is as follows: Condensation gravitates from the heater through check C to the body of the trap, the check W being closed. When the bowl is full enough to overcome the weight of the counterbalance, it sinks and opens up the live-steam valve V. This admits steam to the trap through pipe D, which in turn closes check C and forces the water past the weighted check W to the discharge tank. After the water is discharged the bowl returns to its original position and closes valve V, the weight closes check W, the vent check equalizes the pressure in the bowl and heater, and condensation gravitates to the trap again.

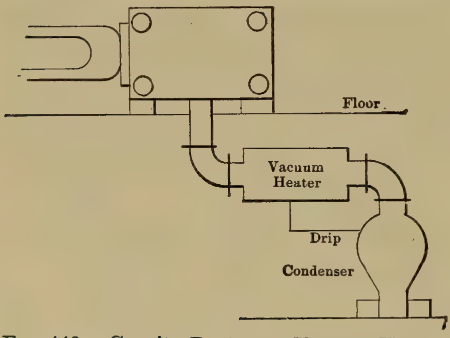


FIG. 449. Gravity Drainage; Vacuum Heater.

359. Drips under Alternate Pressure and Vacuum. — Occasionally the load on an engine is of such a character that the pressure in the receiver alternates from a pressure of 30 or 40 pounds absolute to a vacuum of varying degree. Where the periods of vacuum operation are very few and of short duration, as in the average installation, no attention is paid to the vacuum and the condensation is removed by a trap in the ordinary way. If, however, the periods are of sufficient duration and frequency, the ordinary method is not applicable and the arrangement shown in Fig. 451 may be used. The trap is placed below the receiver

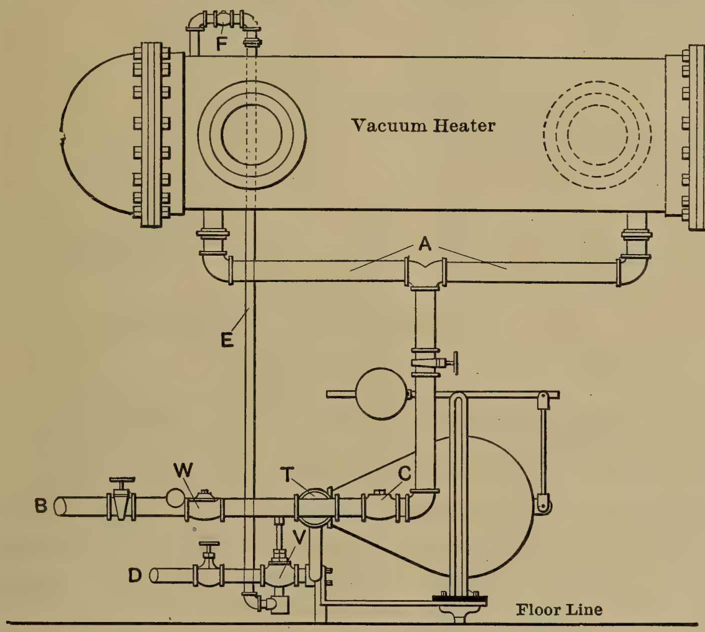


FIG. 450. Method of Draining Heater under Vacuum.

as indicated. The delivery pipe is provided with a weighted check or resistance valve *W* set so as to open outwards from the trap, also a spring water relief valve *R*. Another weighted check *P* is placed in the line leading from the vent to the atmosphere, and a plain check *C* in the line leading back into the receiver. This arrangement of valves permits the venting of the trap after discharge and effectually excludes air from the trap when there is less than atmospheric pressure on the receiver. With the relief valve set to open at a pressure in excess of the maximum receiver pressure it acts as a "stop" in the pipe and the water must enter the trap. When the trap discharges, the live steam supplied through the pipe attached to the steam valve forces the water

through the weighted check and relief valves into the sewer or receiving tank. When working with a vacuum, the pressures in receiver and trap are equalized through the vent connection and the condensation flows into the trap by gravity. The operation of discharge is the same as in the case of pressure.

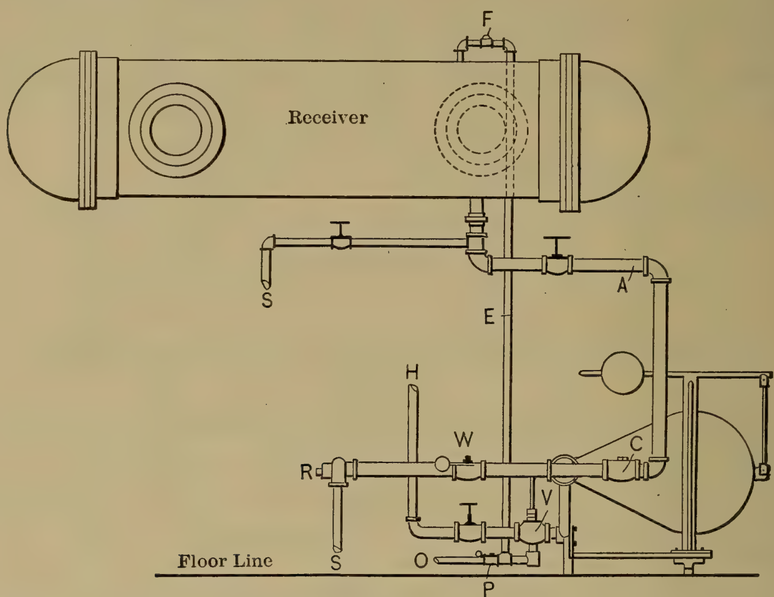


Fig. 451. Method of Draining Receivers under Alternate Vacuum and Pressure.

360. The Steam Loop. — Fig. 452 illustrates the principles of the "steam loop" for automatically returning high-pressure drips to the boiler. In the figure the loop is returning the condensation from a steam separator to a boiler above the level of the separator. The apparatus is very simple, consisting of a horizontal and two vertical lengths of plain pipe placed as indicated. Pipes *R* and *B* may be covered but "horizontal" *A* is left uncovered, as its function is that of a condenser. The operation is as follows: Circulation is first started by opening stop valve *O* at the bottom of the drop leg until steam escapes. The valve is then closed and the steam in the horizontal *A* condenses and gravitates to the drop leg *B*. On account of the slight reduction in pressure in the horizontal a mixture of spray and steam flows from the separator chamber to the horizontal, and, condensing, gravitates to the drop leg. The column of water in the drop leg rises until its static head balances the difference in pressure in the riser *R* and the horizontal. In other words, a decrease in pressure in the horizontal produces similar effects on the contents of the riser and drop leg but in a degree in-

versely proportional to their densities. Any further accumulation causes an equal amount to pass from the bottom of the column to the boiler, since the pressure in the boiler is then less than that at the bottom of the column; that is, the steam pressure on the top of the water column plus the hydrostatic head H is greater than the pressure in the boiler. Once started the process is continuous and requires no further attention.

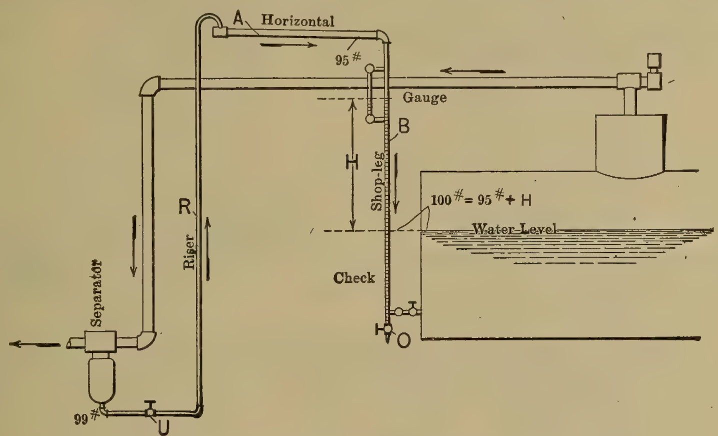


FIG. 452. General Arrangement of the Simple "Steam Loop."

361. The Holly Loop. — In the application of the steam loop where many points requiring drainage are connected to many boilers and conditions are more complex, some method other than the simple one of radiation may be advisable to secure the necessary lower pressure at the top of the loop. Such a method is illustrated in Fig. 453. This arrangement differs from the simple loop in that all condensation first gravitates to a "Holly" receiver (shown in detail in Fig. 454) before passing into the "riser." The receiver is placed below the lowest point to be drained and serves as a storage for large or unusual quantities of water and enables the riser to act at a constant rate independent of variable discharge into the receiver. Furthermore, the lower pressure in the discharge chamber necessary to secure the lifting of the mingled steam and water through the riser, instead of being created by condensation as in the simple loop, is produced by a reducing valve B discharging into the feed-water heater. The operation of the Holly loop is as follows: Circulation is started by opening valve D until steam appears. Valve D is then closed and the reducing valve is put into commission. Condensation from separators, traps, and pipes gravitates to the "receiver," from which it is forced into the "riser" in the form of a spray. The spraying effect is produced by a series of

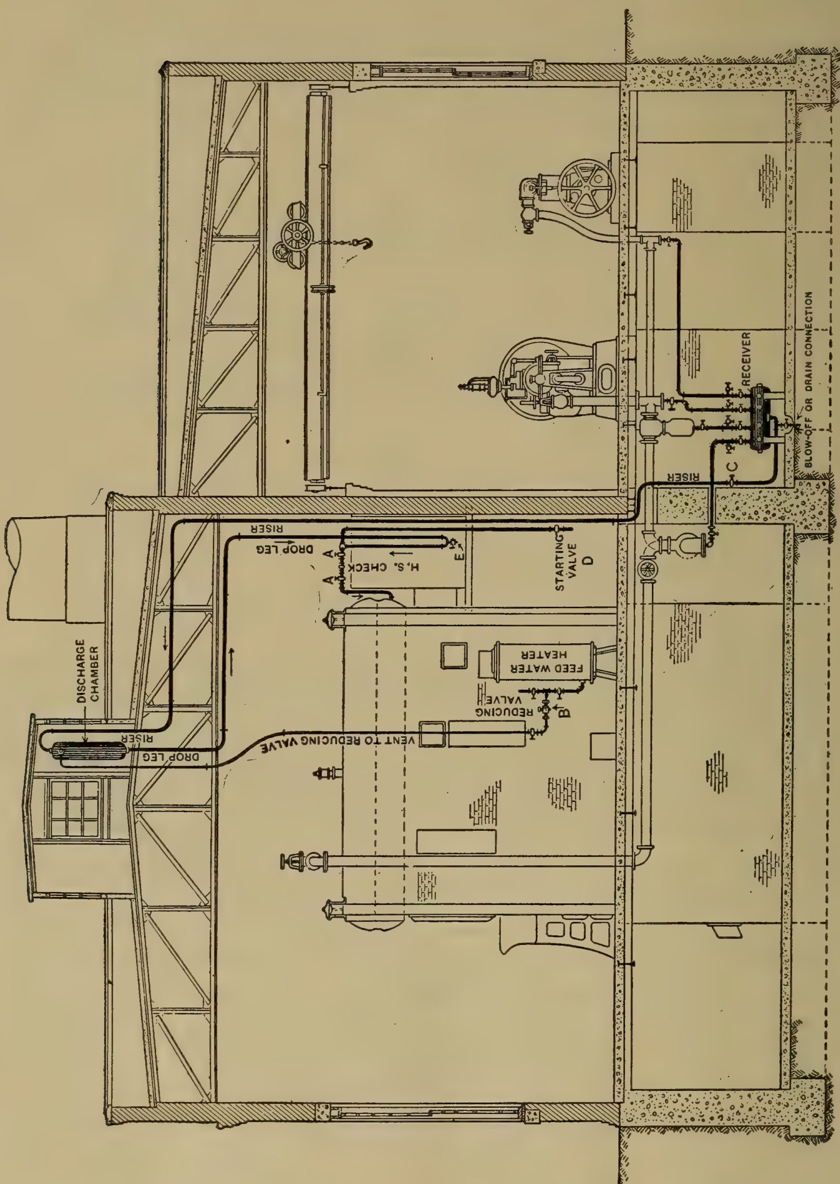


FIG. 453. General Arrangement of the Holly Loop.

holes drilled in pipe A, Fig. 454. From this receiver the spray and moisture rise to the "discharge chamber," on account of the lower pressure at that point, where the steam and entrained water are separated, the water gravitating to the bottom of the chamber and thence to the drop leg, and the steam discharging through the reducing valve into the heater. The principles of operation are exactly the same as in the simple steam loop.

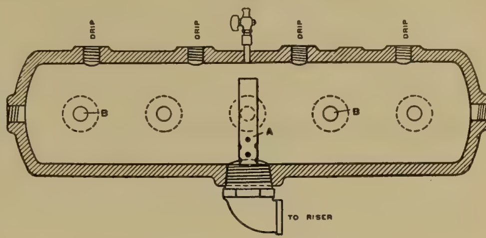


FIG. 454. Holly Receiver.

362. Returns Tank and Pump. — Low-pressure drips in connection with heating systems may be returned to the boiler along with the condensation from the heating system by a combined pump and receiver as shown in Fig. 455. The height of water in the tank controls the

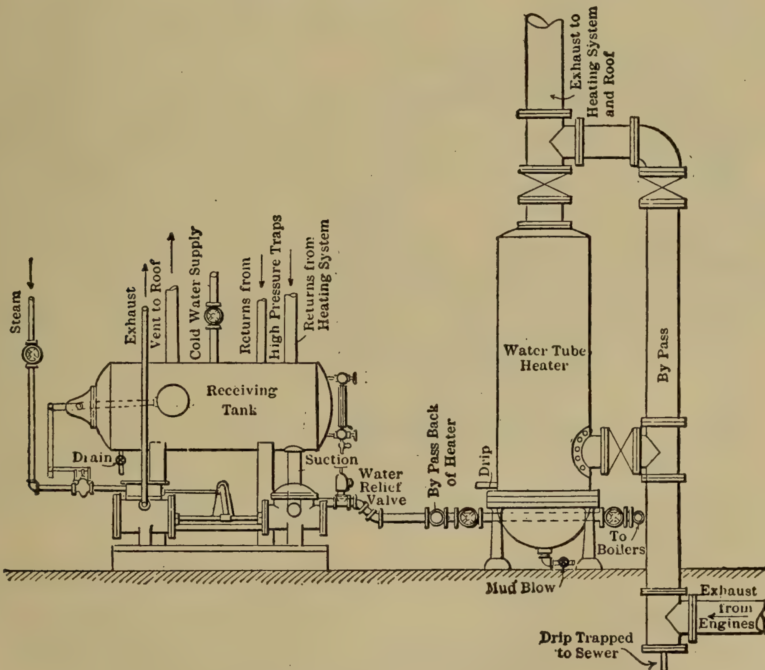


FIG. 455. Returns Tank and Pump.

operation of the pump through the medium of a float and throttle valve. This combination of float and balanced throttle valve is sometimes called a "pump governor." In the illustration the pump forces the

returns through a closed heater before delivering them to the boiler, though they are oftentimes returned directly. The tank is vented to the atmosphere to prevent it from becoming "air bound." The cold-water supply or make-up water is sometimes discharged into the receiving tank as indicated. With open heaters the cold supply is ordinarily controlled by a float within the heater itself.

363. Office Building Drains.—In the power plants of tall office buildings the public sewers are often above the basement level, and it is necessary to remove all liquid wastes mechanically.

The Shone pneumatic ejector has been found to serve this purpose effectually. This apparatus is placed in a pit in the basement floor into which all sewage, drips from engines, washings from boilers, and ground water gravitate, and are automatically discharged into the street sewer by means of compressed air.

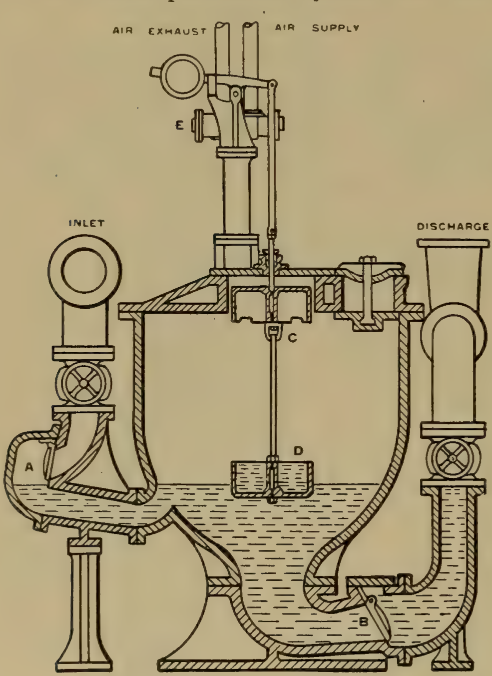


Fig. 456. Shone Ejector.

Fig. 456 gives a sectional view of a Shone ejector of ordinary construction. It consists essentially of a closed vessel furnished with inlet and discharge connections fitted with check valves, *A* and *B*, opening in opposite directions with regard to the ejector. Two cast-iron bells, *C* and *D*, are linked to each other, in reverse positions, the rising and falling of which control the supply of compressed air through the agency of automatic valve *E*.

The bells are shown in their lowest position, the supply of compressed air is cut off from the ejector, and the inside of the vessel is open to the atmosphere. The sewage gravitating into the ejector raises the bell *C*, which in turn actuates the automatic valve *E*, thereby closing the connection between the inside of the ejector and the atmosphere and opening the connection with the compressed air. The air pressure expels the contents through the bell-mouthed opening at the bottom and the discharge valve *B* into the main sewer. Discharge continues until the

level falls to such a point that the weight of the sewage retained in the bell D is sufficient to pull it down, thereby reversing the automatic valve. This cuts off the supply of compressed air and reduces the pressure to that of the atmosphere.

The positions of the bells are so adjusted that compressed air is not admitted until the ejector is full, and is not allowed to exhaust until emptied down to the discharge level; thus the ejector discharges a fixed quantity each time it operates.

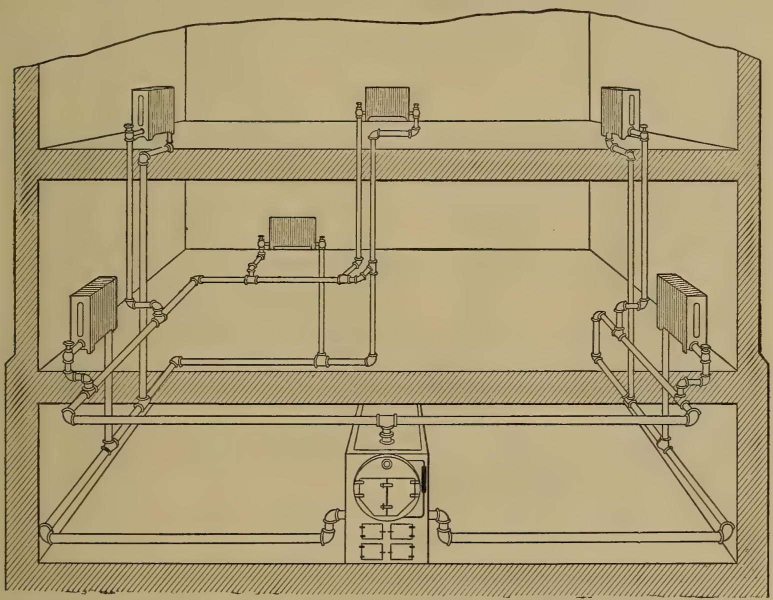


FIG. 457. Radiator Drains, Single Gravity System.

Two ejectors, each of a capacity suitable for handling the average flow of tributary sewage and so arranged that they can work either independently or together, are usually installed at each ejector station.

The main sanitary sewer of the building usually discharges directly into the ejectors, the surface water, drips, etc., being collected in a neighboring sump. The latter is connected to the sanitary sewer through a trap or back-water valve.

364. Radiator Drains. — The condensation from steam heating radiators is invariably drained back to the boiler. In small heating plants with steam pressures of from 1 to 10 pounds gauge pressures, the water of condensation is ordinarily allowed to gravitate directly to the boiler as shown in Fig. 457. In large plants the steam is often circulated below atmospheric pressure, in which case the condensation is withdrawn from

the radiators by mechanical means; see paragraphs 380 and 381. Occasionally small plants are operated below atmospheric pressure. An application of the latter is shown in Fig. 458 and the operation is as follows: Steam is generated in the boiler at from 2 to 5 pounds gauge and, in flowing through the pipes to the radiators, forces the air

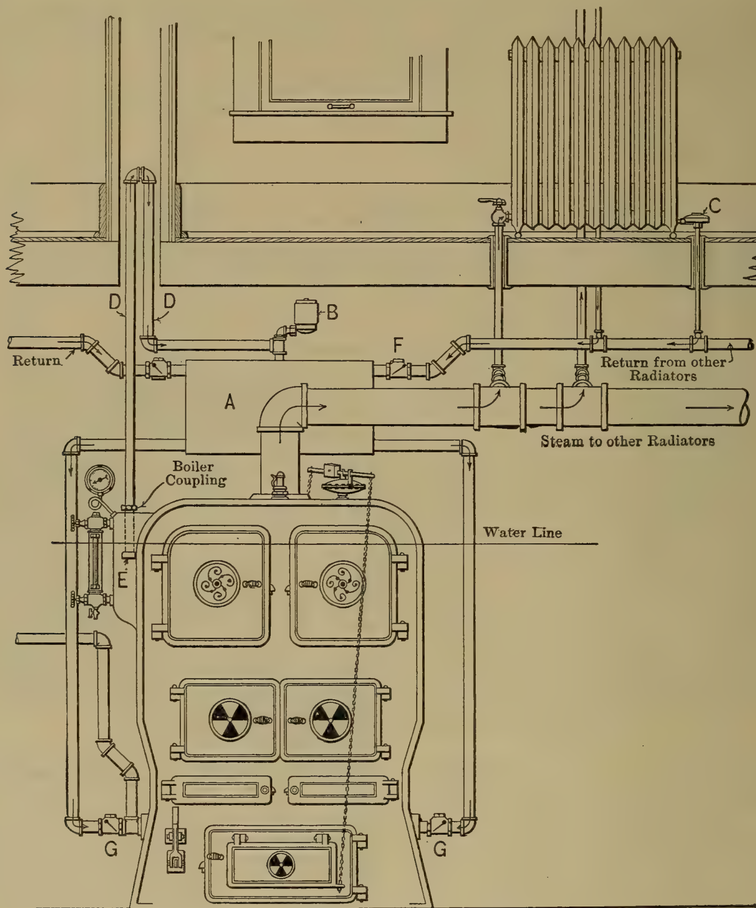


FIG. 458. Radiator Drains, Dunham "Vacuo-Vapor" System.

entrainment before it through the radiator trap into return tank *A*. From the latter it is discharged to the atmosphere through automatic air valve *B*. After the air has been expelled the steam comes into contact with the disk of the radiator trap (see Fig. 442) and the supply is cut off thereby preventing further discharge to tank *A*. Condensation collects in the radiator until its temperature is lowered sufficiently to cause the trap to open and the flow is again established. As soon as

steam strikes the disk of the radiator valve the flow is cut off and the cycle is repeated. The flow of steam to tank A eventually causes the water level in the boiler to fall until it reaches the mouth of the equalizing tube E. As soon as the end of the tube is uncovered steam flows through it into tank A. Steam in tank A immediately closes air valve B and check valve F; the pressure in the tank becomes the same as that in the boiler and the water gravitates to the boiler through check G. The water returning to the boiler raises the water line and seals the equalizing tube. The steam in tank A condenses and forms a vacuum of varying degree over all the return lines. The system is a sealed one and the operation continuous.

CHAPTER XV.

PIPING AND PIPE FITTINGS.

365. General. — The advent of high pressures and superheat is responsible for the elimination of many of the older systems of piping, the tendency being towards greater uniformity in design, particularly in electric central-station work. In isolated stations the conditions of operation and installation are so variable that each case presents an entirely different problem. In any system of piping the fundamental object is to conduct the fluid in the safest and most economical manner.

The material should be the best obtainable and the system so flexible that a break-down in one element will not necessitate the closing down of the entire plant. On the other hand, flexibility increases the number of parts and, unless first cost is of little importance, tends to weaken the system as a whole. It is a safe general proposition to say that the best pipe and fittings, irrespective of first cost, will prove the most economical in the end, but few owners of power plants are willing to take this view.

366. Drawings. — An assembly drawing of the entire installation giving the location of all valves and fittings is necessary in order to avoid interference, and particularly where a number of fittings are to be close together. Detailed drawings should also be provided of each division of the piping to facilitate installation, as, for example, the high-pressure steam, the exhaust steam, the feed water, the condensing water, the oil, the heating, and the sanitary piping. As a rule, lower and more uniform bids will be obtained from an isometric or perspective sketch, as in Fig. 459, than from conventional plan and elevation drawings, due, no doubt, to the greater ease with which the drawing is interpreted. A complete set of specifications for a piping system is given in paragraph 479 and illustrates the usual practice along this line.

367. Materials for Pipes and Fittings. — The following materials are used in the construction of pipes for steam, water, and gases.

	Average Tensile Strength
Low-carbon or mild steel	65,000 lbs. per sq. in.
Wrought iron	50,000 lbs. per sq. in.
Cast iron, high grade	20,000 lbs. per sq. in.
Cast steel	50,000 lbs. per sq. in.
Wrought copper	33,000 lbs. per sq. in.
Brass	18,000 lbs. per sq. in.
Special alloys and compounds	15,000–60,000 lbs. per sq. in.

Mild Steel. — The greater portion of the piping in the average steam power plant is of mild steel, lap or butt welded for high pressures and riveted for very low pressures and large diameters. Steel pipe is considerably cheaper than that manufactured from other material and fulfills practically all requirements for general service.

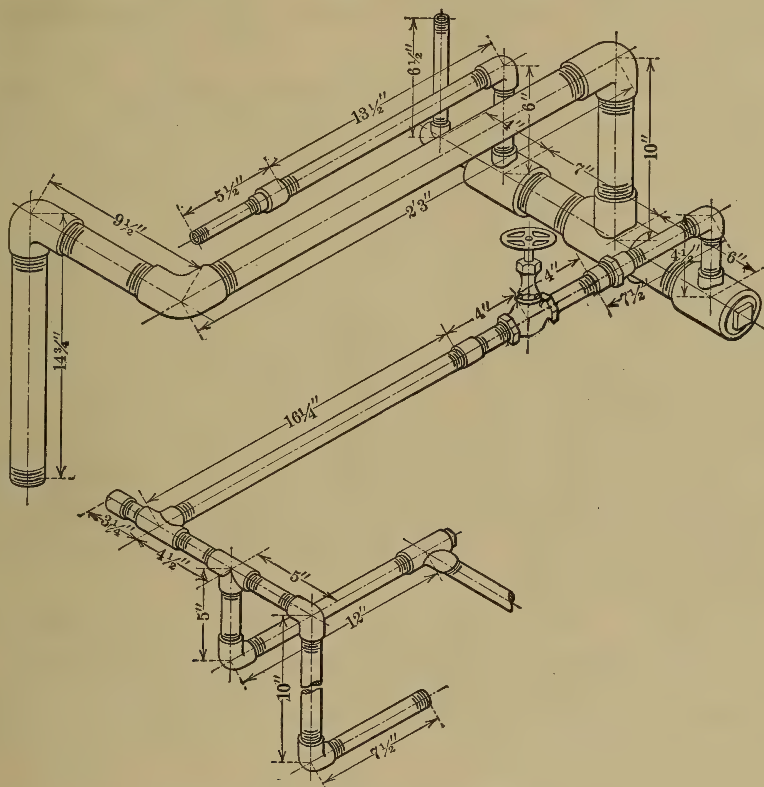


Fig. 459. A Typical Isometric Pipe Drawing.

Wrought Iron. — "Wrought-iron" pipe in a commercial sense refers to mild-steel pipe and unless stress is laid upon the term "puddled iron" mild steel is ordinarily furnished. Puddled-iron pipe is not much in evidence in steam power plant work since mild steel is cheaper and fulfills all requirements. The claims that wrought-iron pipe resists corrosion to a greater extent than mild-steel pipe have not been substantiated in practice. Numerous investigations have been made of late which show that mild steel is equal if not superior to wrought iron in many ways.

Cast-iron Pipes. — Cast iron is little used for high-pressure steam piping except occasionally in the construction of headers where a number of branches lead into a single pipe, in which case the number of joints is greatly reduced and the cost considerably less than for wrought-iron or steel pipe with numerous fittings and joints. The chief objections to cast iron for high-pressure steam are its weight and lack of homogeneity. It is mostly used in connection with water service and sanitation.

Cast-steel Pipe. — Cast-steel headers are sometimes used in power plants for highly superheated steam, since the material is not affected by temperature variations to the same extent as cast iron. High first cost and the difficulty of securing castings free from blowholes have prevented its more general use. (See also paragraph 127.)

Copper Pipes. — Copper steam pipes were in common use for many years in marine service on account of their flexibility. To increase the bursting strength, pipes above 6 inches in diameter were generally wound with a close spiral of copper or composition wire. In recent years wrought-iron and steel pipe bends have practically superseded copper for flexible connections. As a rule the use of copper pipes should be avoided on account of the rapid deterioration of the metal under high temperatures and stress variations. The cost is prohibitive for most purposes and this alone prevents it from being seriously considered in the manufacture of pipe. Copper expansion joints are occasionally used in low-pressure work.

Brass Pipes. — Brass is little used in the construction of pipes on account of its high cost. It withstands corrosive action much better than iron or steel and is often used in connecting the feed main with the boiler drum. Special alloys, nickel steel, "ferrosteel," malleable iron, and the like have been used in the manufacture of pipes, and possess points of superiority over wrought iron and steel for some purposes, as for highly superheated steam, but the cost is prohibitive for average steam power plant practice.

Materials for Fittings. — Elbows, tees, flanges, and similar fittings are usually made of cast iron, malleable iron, or pressed steel, though cast steel, "ferrosteel," and other steel compounds are used to a limited extent. Standard cast-iron fittings are recommended for saturated steam and for pressures of 100 pounds per square inch or less, and extra heavy cast-iron fittings for higher pressures. Malleable-iron fittings are lighter and neater than cast-iron and are extensively used for small sizes of steam and gas pipe. Cast or pressed steel is recommended for very high pressures and superheat.

368. Size and Strength of Commercial Pipe. — Wrought-iron and mild-steel pipes are marketed in standard sizes. Those most commonly used in steam power plants are designated as

1. Merchant or standard pipe.
2. Full-weight pipe.
3. Large O.D. pipe.
4. Extra heavy.
5. Double extra heavy.

Table 107 gives the dimensions of standard "full-weight" pipe, which is specified by the nominal inside diameter up to and including 12 inches and based on the Briggs' standard. Pipes larger than 12 inches are designated by the actual outside diameter (O.D.), and are made in various weights as determined by the thickness of metal specified. Manufacturers specify that "full-weight" pipe may have a variation of 5 per cent above or 5 per cent below the nominal or table weights, but merchant pipe, which is the standard pipe of commerce, such as manufacturers and jobbers usually carry in stock, is almost invariably under the nominal weight. It varies somewhat among the different mills, but usually lies between 5 and 10 per cent under the table weight. The smaller sizes of merchant pipe, $\frac{1}{8}$ inch to 3 inches, are butt-welded and the larger sizes are lap-welded.

Extra heavy and double extra heavy pipe have the same external diameter as the standard, but are of greater thickness and hence the internal diameter is smaller. Taking the thickness of the standard pipe as 1, that of the extra heavy is approximately 1.4 and of the double extra heavy 2.8.

Wrought-iron and steel pipes are ordinarily designed with factors of safety of from 6 to 15, with an average not far from 10. The standard hydrostatic tests to which the various pipes are subjected at the mills are as follows:

	Hydrostatic Pressure, Lbs. per Sq. In.
Standard, butt-welded, $\frac{1}{8}$–3 in.	600 to 1,000
Standard, lap-welded, 3–12 in.	500 to 1,000
Extra heavy, butt-welded, $\frac{1}{8}$–3 in.	600 to 1,500
Extra heavy, lap-welded, $1\frac{1}{2}$–12 in.	600 to 1,500
Double extra heavy, butt-welded, $\frac{1}{8}$–$2\frac{1}{2}$ in	600 to 1,500
Double extra heavy, lap-welded, $1\frac{1}{2}$–8 in	1,200 to 1,500

The pressure necessary to burst piping is far above anything likely to occur in ordinary practice on account of the thickness of material necessary to permit of threading. (See Table 106.)

Riveted Pipes. — For low pressures and large diameters, pipes are constructed of thin sheets of boiler steel with riveted joints, the seams

being either longitudinal and circumferential, or spiral. Such pipes are not necessarily limited to large sizes and low pressures, though this is the usual practice.

Pipe fittings are classed as screwed or flanged.

TABLE 106.

BURSTING PRESSURE OF "STANDARD" MILD-STEEL PIPE.**

No. of Specimen.	Nominal Diameter, Inches.	Actual Bursting Pressure, Lbs. per Sq. In.	No. of Specimen.	Nominal Diameter, Inches.	Actual Bursting Pressure, Lbs. per Sq. In.
†1	1	7800	‡7	3	3500
†2	1	7700	‡8	3	3500
†3	1	7700	†9	3	3000
		Average 7730			Average 3330
†4	2	4950	§10	4	1800
†5	2	4800	§11	4	1700
‡6	2	5500			Average 1750
		Average 5080	§12	5	2500
			§13	5	2600
					Average 2550
			§14	6	3200

* Tests made at Armour Institute of Technology.

Specimens were taken at random from a lot of new pipe; length of test specimens, 5 ft. Specimens threaded at both ends and capped.

† Failed at weld. ‡ Failed in body of pipe. § Failed at threaded end.

369. Screwed Fittings, Pipe Threads. — For screw connections the ends of pipes and fittings are threaded to conform to the Briggs or United States standard system, as shown in Fig. 460. The end of the

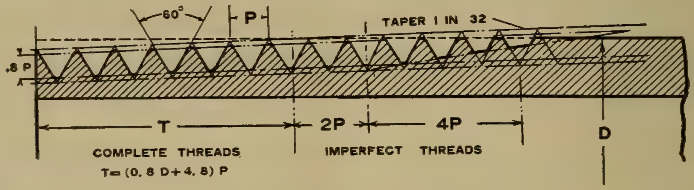


FIG. 460. Standard U. S. Pipe Thread.

pipe is tapered 1 to 32 with the axis, the angle of the thread being 60 degrees and slightly rounded at top and bottom. The proper length of perfect threads is given by the formula

$$T = \frac{(0.8\,D + 4.8)}{n}, \tag{253}$$

in which

T = length in inches.

D = actual external diameter of the tube, inches.

n = number of threads per inch.

TABLE 107.

STANDARD DIMENSIONS OF WROUGHT-IRON AND STEEL STEAM, GAS, AND WATER PIPE.

Diameter.			Nominal Thickness.	Circumference.		Transverse Areas.			Length of Pipe per Square Foot of		Length of Pipe Containing One Cubic Foot.	Nominal Weight per Foot.	Number of Threads per Inch of Screw.
Nominal Internal.	Actual External Diameter.	Approximate Internal Diameter.		External.	Internal.	External.	Internal.	Metal.	External Surface.	Internal Surface.			
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Sq. Inch.	Sq. Inch.	Sq. Inch.	Feet.	Feet.	Feet.	Pounds.	
1/8	.405	.27	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513	.241	27
1/4	.54	.364	.088	1.696	1.144	.229	.1041	.1249	7.075	10.49	1383.3	.42	18
3/8	.675	.494	.091	2.121	1.552	.358	.1917	.1663	5.657	7.73	751.2	.559	18
1/2	.84	.623	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.837	14
3/4	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270	1.115	14
1	1.315	1.048	.134	4.131	3.292	1.358	.8626	.4954	2.904	3.645	166.9	1.668	11 1/2
1 1/4	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.244	11 1/2
1 1/2	1.9	1.611	.145	5.969	5.061	2.835	2.038	.797	2.01	2.371	70.66	2.678	11 1/2
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.609	11 1/2
2 1/2	2.875	2.468	.204	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1	5.739	8
3	3.5	3.067	.217	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.536	8
3 1/2	4	3.548	.226	12.566	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.001	8
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.665	8
4 1/2	5	4.508	.246	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.49	8
5	5.563	5.045	.259	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.502	8
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.98	18.762	8
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	23.271	8
8	8.625	7.982	.322	27.096	25.076	58.426	50.04	8.386	.443	.478	2.88	28.177	8
9	9.625	8.937	.344	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.701	8
10	10.75	10.019	.366	33.772	31.477	90.763	78.839	11.924	.355	.382	1.82	40.065	8
11	11.75	11	....	36.914	34.558	108.434	95.033	13.401	.325	.347	1.51	45.028	8
12	12.75	12	....	40.055	37.7	127.677	113.098	14.579	.299	.319	1.27	48.985	8

The imperfect portion of the thread is simply incidental to the process of cutting. The object of the taper is to facilitate "taking hold" in making up the joint. Table 107 gives the number of threads per inch for various sizes of standard pipe. When properly constructed a screwed joint will hold against any pressure consistent with the strength of the pipe. For example, the ultimate bursting strength of a "standard" 2-inch pipe is about 5000 pounds per square inch, while the stripping strength of the joint (with perfect threads) is 225,000 pounds.

TABLE 108.

STANDARD BOILER TUBES.

Table of Standard Dimensions.

Diameter.		Standard Thickness.		Transverse Areas.		Area of Surface per Foot of Tube.		Nominal Weight per Foot — Lbs.				
External.	Internal.	Nearest B.W.G.		External.	Internal.	External.	Internal.	Standard Thickness.	One Extra Wire Gauge.	Two Extra Wire Gauges.	Three Extra Wire Gauges.	Four Extra Wire Gauges.
Ins.	Ins.	No.	Ins.	Sq. In.	Sq. In.	Sq. Ft.	Sq. Ft.					
1	0.810	13	.095	0.785	0.515	.262	.212	0.90	1.04	1.13	1.24	1.35
$1\frac{1}{4}$	1.060	13	.095	1.227	0.882	.327	.277	1.15	1.33	1.45	1.60	1.74
$1\frac{1}{2}$	1.310	13	.095	1.767	1.348	.392	.343	1.40	1.62	1.77	1.96	2.14
$1\frac{3}{4}$	1.560	13	.095	2.405	1.911	.458	.408	1.66	1.91	2.09	2.31	2.53
2	1.810	13	.095	3.142	2.573	.523	.474	1.91	2.20	2.41	2.67	2.93
$2\frac{1}{4}$	2.060	13	.095	3.976	3.333	.589	.539	2.16	2.49	2.73	3.03	3.32
$2\frac{1}{2}$	2.282	12	.109	4.909	4.090	.654	.597	2.75	3.05	3.39	3.72	4.12
$2\frac{3}{4}$	2.532	12	.109	5.940	5.035	.720	.663	3.04	3.37	3.74	4.11	4.56
3	2.782	12	.109	7.069	6.079	.785	.728	3.33	3.69	4.10	4.51	5.00
$3\frac{1}{4}$	3.010	11	.120	8.296	7.116	.851	.788	3.96	4.46	4.90	5.44	5.90
$3\frac{1}{2}$	3.260	11	.120	9.621	8.347	.916	.853	4.28	4.82	5.30	5.88	6.38
$3\frac{3}{4}$	3.510	11	.120	11.045	9.676	.982	.919	4.60	5.18	5.69	6.32	6.86
4	3.732	10	.134	12.566	10.939	1.047	.977	5.47	6.09	6.76	7.34	8.23
$4\frac{1}{2}$	4.232	10	.134	15.904	14.066	1.178	1.108	6.17	6.88	7.64	8.31	9.32
5	4.704	9	.148	19.635	17.379	1.309	1.231	7.58	8.52	9.27	10.40	11.23
6	5.670	8	.165	28.274	25.250	1.571	1.484	10.16	11.19	12.57	13.58	14.65

The threads, however, are often poorly cut and the parts screwed together improperly cleaned and lubricated, thus causing leakage between the threads.

370. Flanged Fittings. — In cast-iron pipes, valves, tees, and other fittings the flange is always a part of the casting, but for joining the two ends of a steel or wrought-iron pipe the flanges may be fastened to the pipe in a number of ways. Fig. 461, *A* to *H*, illustrates methods most commonly used. In *A* to *C* the pipes are screwed into cast-iron or forged-steel flanges and the two faces, with metallic or composition gasket between, are drawn together by bolts. *A* illustrates the most

common and inexpensive of flanged joints, which requires no special tools and can be made up at the place of erection. It gives satisfactory results for pressures of 100 pounds or less, but for higher pressures leakage is apt to take place between the threads. The flanges are

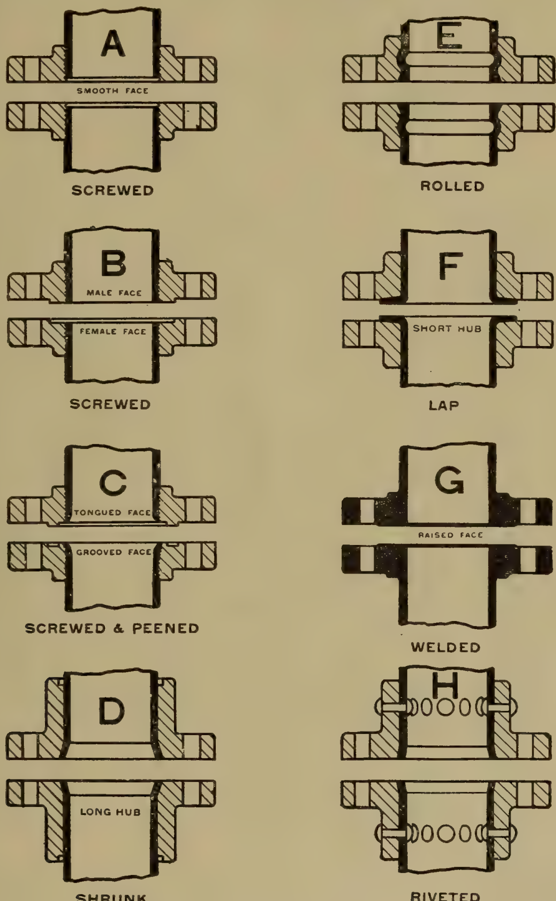


FIG. 461. Types of Pipe Flanges.

sometimes made with a long thread and a recess which can be calked with soft metal. A similar joint is made with the pipe screwed beyond the face of the flange and the two faced off together, either plane or as shown in *B*, which is known as a *male and female* or *hydraulic* joint. This method forms a very reliable joint, since the ends of the pipe bear on the gasket, and the gasket is prevented from being blown out. An objection lies in the difficulty of opening the line to remove the gasket or replace a fitting. *C* is a modification known as the

tongued and grooved joint, which uses an extremely narrow gasket. Such flanges may be subjected to severe strains when the bolts are drawn up, owing to the small area of contact. Corrugated copper or steel gaskets are recommended, since soft material is apt to be squeezed out. In *C* the ends of the pipe are *peened*, which is an improvement

TABLE 109.

DIMENSIONS OF CAST-IRON PIPE.*

Nominal Inside Diameter, Inches.	Standard Thickness and Weight.								
	Class A. 100 Feet Head. 43 Pounds Pressure.			Class B. 200 Feet Head. 86 Pounds Pressure.			Class C. 300 Feet Head. 130 Pounds Pressure.		
	Thickness, Inches.	Weight per Foot.	Weight per Length.	Thickness, Inches.	Weight per Foot.	Weight per Length.	Thickness, Inches.	Weight per Foot.	Weight per Length.
4	.42	20.0	240	.45	21.7	260	.48	23.3	280
6	.44	30.8	370	.48	33.3	400	.51	35.8	430
8	.46	42.9	515	.51	47.5	570	.56	52.1	625
10	.50	57.1	685	.57	63.8	765	.62	70.8	850
12	.54	72.5	870	.62	82.1	985	.68	91.7	1,100
14	.57	89.6	1,075	.66	102.5	1,230	.74	116.7	1,400
16	.60	108.3	1,300	.70	125.0	1,500	.80	143.8	1,725
18	.64	129.2	1,550	.75	150.0	1,800	.87	175.0	2,100
20	.67	150.0	1,800	.80	175.0	2,100	.92	208.3	2,500
24	.76	204.2	2,450	.89	233.3	2,800	1.04	279.2	3,350
30	.88	291.7	3,500	1.03	333.3	4,000	1.20	400.0	4,800
36	.99	391.7	4,700	1.15	454.2	5,450	1.36	545.8	6,550
42	1.10	512.5	6,150	1.28	591.7	7,100	1.54	716.7	8,600
48	1.26	666.7	8,000	1.42	750.0	9,000	1.71	908.3	10,900
54	1.35	800.0	9,600	1.55	933.3	11,200	1.90	1141.7	13,700
60	1.39	916.7	11,000	1.67	1104.2	13,250	2.00	1341.7	16,100
72	1.62	1283.4	15,400	1.95	1545.8	18,550	2.39	1904.2	22,850
84	1.72	1633.4	19,600	2.22	2104.2	25,250	......	......	....

* Adopted standards of Am. Water W'ks Ass'n. The above weights are per length to lay 12 feet, including standard sockets; proportionate allowance to be made for any variation. All weights are approximate.

Divisions of Riveted Steel Pipes: Power, March 7, 1911, p. 377.

over the simple screwed joint. *D* illustrates a *shrunk* joint. The flanges are bored for a shrink fit and forced over the pipe when at a red heat. After cooling the end is beaded over into a recess on the face of the flange and a light cut taken from both. *H* shows a modification in which the hub is riveted to the pipe. *E* illustrates a joint constructed by rolling the pipe into a corrugation in the flange. The end of the pipe is then faced off flush.

One of the best commercial joints is illustrated by F and is known as the *lap joint*. The pipe is expanded as indicated and a light cut is then taken from the flared ends to insure a tight joint. The flanges are loose and permit of considerable flexibility in shifting them through various angles. This is sometimes called the *Van Stone* joint.

Pipes with flanges *welded* on the end as in G have proved the most reliable of all and though costly are considered the standard for high-pressure and high-temperature work. The faces are ordinarily raised $\frac{1}{32}$ to $\frac{1}{16}$ inch inside the bolt holes and ground to a steam-tight fit, so that thick gaskets are unnecessary.

For moderately high pressures and temperatures any of the joints when well made will prove satisfactory. For extremely high pressures and temperatures the lap or welded joints are preferable.

The comparative costs of various flanges are given in Table 112.

Table 110 gives the dimensions of standard and extra-heavy flanges and fittings as adopted July 10, 1912, by manufacturers, and Table 111 the dimensions adopted by "The Societies," Oct. 25, 1911. Since neither of these two standards has been universally adopted in this country they should be used with caution.

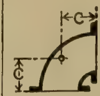
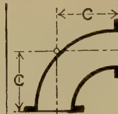
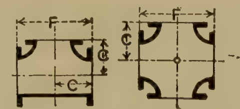
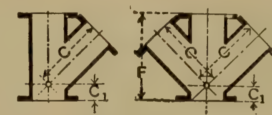
The following explanatory notes refer to tables 110 and 111. The societies' notes appear in Roman type and the manufacturers' variations, wherever they occur, follow in *italics:*

1. Standard or extra-heavy reducing elbows carry the same dimensions center to face as the regular elbows of the largest straight size.
2. Standard or extra-heavy tees, crosses, and laterals, reducing on run, carry the same dimensions face to face as the largest straight size.
3. If flanged fittings for lower working pressures than 125 pounds are made, they shall conform in all dimensions, except in thickness of shell, to this standard, and shall have the guaranteed working pressure cast on each fitting. Flanges for these fittings must be of standard dimensions.
4. Where long-turn fittings are specified, it has reference only to elbows, which are made in two center-to-face dimensions, to be known as "elbows" and "long-turn" elbows, the latter being used only when so specified.
5. All standard-weight fittings must be guaranteed for 125 pounds and extra-heavy fittings for 250 pounds working pressure, and each fitting must have some mark cast on it indicating the maker and the guaranteed working steam pressure.
6. All extra-heavy fittings and flanges to have a raised surface $\frac{1}{16}$ inch high inside of the bolt holes for the gasket.

Standard-weight fittings and flanges to be plain faced.

TABLE 110. "MANUFACTURERS 1912 SCHEDULE" OF FLANGES AND FLANGED FITTINGS.

(Adopted July 10, 1912.) Standard-weight Flanged Fittings.

Size, Inches.	All Fittings and Flanges. Diameter of Flange.	Thickness of Flange.	Number of Bolts.	Diameter of Bolt Circle.	Diameter of Bolt Holes.	Size of Bolts.	90° Elbow.  Center to Face "C."	45° Elbow.  Center to Face "C."	Long Turn Elbow.  Center to Face "C."	Tees and Crosses.  Center to Face "C."	Tees and Crosses. Center to Face Outlet 8" and Under.	Tees and Crosses. Face to Face "F"	Laterals and Y Branches.  Center to Face "C."	Laterals and Y Branches. Center to Face on Short End "C."	Laterals and Y Branches. Center to Face of Outlets Smaller in Dia.	Laterals and Y Branches. Face to Face "F."	Reducer.  Face to Face "F."
1	4	7/16	4	3	9/16	7/16	3½	2 1¾	5½ 5	3½		7	6	2		8	
1¼	4½	½	4	3⅜	9/16	7/16	3¾	2	6 5½	3¾		7½	6¼	2¼		8½	
1½	5	9/16	4	3⅞	⅝	½	4	2¼	6½ 6	4		8	6⅞	2⅜		9¼	
2	6	⅝	4	4¾	¾	⅝	4½	2½	7 6½	4½		9	8	2½		10½	
2½	7	11/16	4	5½	¾	⅝	5	2¾ 3	7½ 7	5		10	9½	2½		12	
3	7½	¾	4	6	¾	⅝	5½	3	8 7¾	5½		11	10	3		13	6
3½	8½	13/16	4	7	¾	⅝	6	3½	9 8½	6		12	11½	3		14½	6¼
4	9	15/16	8	7½	⅞ ¾	¾ ⅝	6½	4	10 9	6½		13	12	3		15	7
							Outlet 2½ in. and under							2	11	13	
4½	9¼	15/16	8	7¾	⅞	¾	7	4¼ 4	11 9¼	7		14	12½	3	11⅞*	15½	7½
							Outlet 2½ in. and under							2	11	13	
5	10	15/16	8	8½	⅞	¾	7½	4½	12 10¼	7½		15	13½	3½	12¼*	17	8
							Outlet 3 in. and under							2	12	14	
6	11	1	8	9½	⅞	¾	8	5	13 11½	8		16	14½	3½	13*	18	9
							Outlet 3 in. and under							1½	13½	15	
7	12½	1 1/16	8	10¾	⅞	¾	8½	5½	14½ 12¾	8½		17	16½	4	13¾*	20½	10
							Outlet 3½ in. and under							1½	15	16	
8	13½	1⅛	8	11¾	⅞	¾	9	6 5½	16 14	9		18	17½	4½	14⅜*	22	11
							Outlet 4 in. and under							1½	15½	16	
9	15	1⅛	12	13¼	⅞	¾	10	6½ 6	18 15¼	10	9	20	19½	4½	17⅞†	24	11½
							Outlet 4½ in. and under							1½	16½	17	
10	16	1 3/16	12	14¼	1	⅞	11	7 6½	20 16½	11	10	22	20½	5	18⅝†	25½	12
							Outlet 5 in. and under							1	18	18	
							Outlet 6 in. and under				9½	18					
12	19	1¼	12	17	1	1⅛	12	7½	22 19	12	11½	24	24½	5½	20†	30	14
							Outlet 6 in. and under							1	20½	20	
							Outlet 8 in. and under				11	20					
14	21	1⅜	14	18¾	1⅛	1	14	7½	24 21½	14	12½	28	27	6	21½†	33	16
							Outlet 7 in. and under							1	23	22	
							Outlet 9 in. and under				13	22					
15	22¼	1⅜	16	20	1⅛	1	14½	8	26 22¾	14½	13	29	28½	6	22⅛†	34½	17
							Outlet 7 in. and under							1	24	23	
							Outlet 9 in. and under				13½	23					
16	23½	1 7/16	16	21¼	1⅛	1	15	8	28 24	15	13½	30	30	6½	22⅞†	36½	18
							Outlet 8 in. and under							1	25½	24	
							Outlet 10 in. and under				14	24					
18	25	1 9/16	16	22¾	1¼	1⅛	16½	8½	30 26½	16½	14½	33	32	7	24¼†	39	19
							Outlet 9 in. and under							1	27½	26	
							Outlet 12 in. and under				15½	26					

20	27½		1 1/16	20	25		1¼		1⅛		18		9½		32	29	18		15½	36		35		8	25¾†	43		20
											Outlet 10 in. and under													1	29½	28		
											Outlet 14 in. and under								17	28								
22	29½		1 3/16	20	27¼		1¼		1⅛		20		10		34	31½	20		16½	40		37½		8½	27⅝†	46		22
											Outlet 10 in. and under													½	31½	29		
											Outlet 15 in. and under								18	28								
24	32		1¼	20	29½		1¼		1⅛		22		11		36	34	22		17½	44		40½		9	28¾†	49½		24
											Outlet 12 in. and under													½	34½	32		
											Outlet 16 in. and under								19	30								

Extra-heavy Flanged Fittings.

1	4½		11/16	4	3¼		⅝		½		4¼	4	2½	2	5½	5	4¼	4		8½	8	7½		2⅝		9¾		
1¼	5		¾	4	3¾		⅝		½		4¼		2½		6	5½	4¼			8½		7½		2⅝		9¾		
1½	6		13/16	4	4½		¾		⅝		4½		2¾		6½	6	4½			9		8½		2½		11		
2	6½		⅞	4	5		¾		⅝		5		3		7	6½	5			10		9½	9	2½		12	11½	
2½	7½		1	4	5⅞		⅞		¾		5½		3¼	3½	7½	7	5½			11		10½		2½		13		
3	8¼		1⅛	8	6⅝		⅞		¾		6		3½		8	7¾	6			12		11½	11	3		14½	14	6
3½	9		1 3/16	8	7¼		⅞		¾		6½		4		9	8½	6½			13		12½		3		15½		6½
4	10		1¼	8	7⅞		⅞		¾		7		4½	4½	10	9	7			14		13½		3		16½		7
											Outlet 2½ in. and under													2	13	15		
4½	10½		1 5/16	8	8¼		⅞		¾		7½		4½		11	9½	7½			15		14½		3½	13⅝*	18		7½
											Outlet 2½ in. and under													2½	13½	15		
5	11		1⅜	8	9¼		⅞		¾		8		5		12	10¼	8			16		16	15	3½	14*	19½	18½	8
											Outlet 3 in. and under													2½	14½	16		
6	12½		1 7/16	12	10⅝		⅞		¾		9	8½	5½		13	11½	9	8½		18	17	17½		4	14⅝*	21½		9
											Outlet 3 in. and under													2½	15½	17		
7	14		1½	12	11⅞		1		⅞		9½	9	5¾	6	14½	12¾	9½	9		19	18	19	18½	4½	15⅝*	23½	23	10
											Outlet 3½ in. and under													2½	16½	18		
8	15		1⅝	12	13		1		⅞		10½	10	6		16	14	10½	10		21	20	20½	20	5	16*	25½	25	11
											Outlet 4 in. and under													2½	18½	20		
9	16¾	16¼	1¾	12	14		1⅛		1		11	10½	6½		18	15¼	11	10½	10½	22	21	22¾	21	5	20⅞†	27¾	26	11½
											Outlet 4½ in. and under													2½	19½	21		
10	18¼	17½	1⅞	16	15¾	15¼	1⅛		1		12	11½	7		20	16½	12	11½	11	24	23	25¼	23	5½	21⅝†	30¾	28½	12
											Outlet 5 in. and under													2½	21½	23		
											Outlet 6 in. and under								11	18								
12	20¾	20	2	16	17¾		1¼	1⅛	1⅛	1	13½	13	8		22	19	13½	13	12	27	26	27¾	26½	6	23†	33¾	32½	14
											Outlet 6 in. and under													2½	24½	26		
											Outlet 8 in. and under								12½	21								
14	23½	22½	2⅛	20	20¼	20	1⅜	1⅛	1¼	1	15	14½	9½	8	24	21½	15	14½	13	30	29	31¼	30	6½	24⅜†	37¾	36½	16
											Outlet 7 in. and under													2½	27½	29		
											Outlet 9 in. and under								13½	22								
15	25	23½	2 3/16	20	21½	21	1⅜	1¼	1¼	1⅛	15½	15	10	8½	26	22¾	15½	15	13½	31	30	33¼	31½	6½	25⅝†	39¾	38	17
											Outlet 7 in. and under													2½	28½	30		
											Outlet 9 in. and under								13½	23								
16	26	25	2¼	20	22½		1½	1¼	1⅜	1⅛	16½	16	10½	9	28	24	16½	16	14	33	32	34½	33	7½	25⅞†	42	40½	18
											Outlet 8 in. and under													3	30½	32		
											Outlet 10 in. and under								15	24								
18	28½	27	2⅜	24	24¾	24½	1½	1¼	1⅜	1⅛	17¾	17	11	9½	30	26½	17¾	17	15	35½	34	37¾	36	8	27¼†	45¾	44	19
											Outlet 9 in. and under													3	32½	34		
											Outlet 12 in. and under								16½	27								
20	31	29½	2½	24	27	26¾	1⅝	1⅜	1½	1¼	19¼	18½	11½	10	32	29	19¼	18½	16	38½	37	40¾	38	8½	28⅝†	49¼	46½	20
											Outlet 10 in. and under													3	36	37		
											Outlet 14 in. and under								17½	30								
22	33	31½	2⅝	28	29¼	28¾	1⅝	1⅜	1½	1¼	20½	20	12	10½	34	31½	20½	20	17	41	40	43¾	41½	9½	30†	53¼	51	22
											Outlet 10 in. and under													3	39	40		
											Outlet 15 in. and under								18½	30								
24	36	34½	2¾	28	32	31¼	1¾	1½	1⅝	1⅜	22½	22	13	11½	36	34	22½	22	18	45	44	47½	44½	10	31½†	57½	54½	24
											Outlet 12 in. and under													3	43	44		
											Outlet 15 in. and under								19½	30								

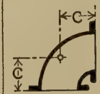
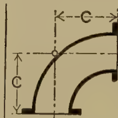
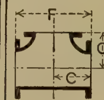
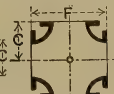
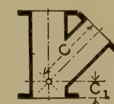
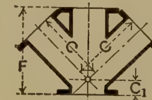
* Outlets 4 in. and under. † Outlets 8 in. and under.

TABLE 111.

"U. S. 1912 STANDARD" SCHEDULE OF FLANGES AND FLANGED FITTINGS.

(Adopted Oct. 25, 1911, by a Committee of the National Association of Master Steam and Hot Water Fitters and of the American Society of Mechanical Engineers.)

Standard-weight Flanged Fittings.

	All Fittings and Flanges.						 90° Elbow.	 45° Elbow.	 Long Turn Elbow.	 Tee.	 Cross.	 Lateral.		 Y Branch.		 "Y."	
Size, Inches.	Diameter of Flange.	Thickness of Flange.	Number of Bolts.	Diameter of Bolt Circle.	Diameter of Bolt Holes.	Size of Bolts.	Center to Face "C."	Center to Face "C."	Center to Face "C."	Center to Face "C."	Face to Face "F."	Center to Face "C."	Center to Face on Short End "C_1."	Center to Face of Outlets.	Face to Face "F."	Center to Face "C."	Center to Face Short End "C_1."
1	4	7/16	4	3	9/16	7/16	3½	2	5½	3½	7	6	2	....	8	3½	2
1¼	4½	½	4	3⅜	9/16	7/16	3¾	2	6	3¾	7½	6¼	2¼	....	8½	3¾	2¼
1½	5	9/16	4	3⅞	⅝	½	4	2¼	6½	4	8	6⅞	2⅜	....	9¼	4	2⅜
2	6	⅝	4	4¾	¾	⅝	4½	2½	7	4½	9	8	2½	....	10½	4½	2½
2½	7	11/16	4	5½	¾	⅝	5	2¾	7½	5	10	9½	2½	....	12	5	2½
3	7½	¾	4	6	¾	⅝	5½	3	8	5½	11	10	3	....	13	5½	3
3½	8½	13/16	4	7	¾	⅝	6	3½	9	6	12	11½	3	....	14½	6	3
4	9	15/16	8	7½	⅞	¾	6½	4	10	6½	13	12	3	....	15	6½	3
4½	9¼	15/16	8	7¾	⅞	¾	7	4¼	11	7	14	12½	3	11⅞	15½	7	3
5	10	15/16	8	8½	⅞	¾	7½	4½	12	7½	15	13½	3½	12¼	17	7½	3½
6	11	1	8	9½	⅞	¾	8	5	13	8	16	14½	3½	13	18	8	3½
7	12½	1 1/16	8	10¾	⅞	¾	8½	5½	14½	8½	17	16½	4	13¾	20½	8½	4
8	13½	1⅛	8	11¾	⅞	¾	9	6	16	9	18	17½	4½	14⅜	22	9	4½
9	15	1⅛	12	13¼	⅞	¾	10	6½	18	10	20	19½	4½	17⅞	24	10	4½
10	16	1 3/16	12	14¼	1	⅞	11	7	20	11	22	20½	5	18⅝	25½	11	5
12	19	1¼	12	17	1	⅞	12	7½	22	12	24	24½	5½	20	30	12	5½
14	21	1⅜	14	18¾	1⅛	1	14	7½	24	14	28	27	6	21½	33	14	6
15	22¼	1⅜	16	20	1⅛	1	14½	8	26	14½	29	28½	6	22⅛	34½	14½	6
16	23½	1 7/16	16	21¼	1⅛	1	15	8	28	15	30	30	6½	22⅞	36½	15	6½
18	25	1 9/16	16	22¾	1¼	1⅛	16½	8½	30	16½	33	32	7	24¼	39	16½	7
20	27½	1 11/16	20	25	1¼	1⅛	18	9½	32	18	36	35	8	25¾	43	18	8
22	29½	1 13/16	20	27¼	1¼	1⅛	20	10	34	20	40	37½	8½	27⅛	46	20	8½
24	32	1⅞	20	29½	1¼	1⅛	22	11	36	22	44	40½	9	28½	49½	22	9
26	34¼	2	24	31¾	1⅜	1¼	23	13	39	23	46	43½	9½	30¾	53	23	9½
28	36½	2 1/16	28	34	1⅜	1¼	24	14	42	24	48	46½	10½	33	57	24	10½
30	38¾	2⅛	28	36	1½	1⅜	25	15	45	25	50	49½	11	35½	60½	25	11

Extra-heavy Flanged Fittings.

1	4½	11/16	4	3¼	⅝	½	4¼	2½	5½	4¼	8½	7⅛	2⅝	....	9¾	4¼	2⅝
1¼	5	¾	4	3¾	⅝	½	4¼	2½	6	4¼	8½	7⅛	2⅝	....	9¾	4¼	2⅝
1½	6	13/16	4	4½	¾	⅝	4½	2¾	6½	4½	9	8½	2½	....	11	4½	2½
2	6½	⅞	4	5	¾	⅝	5	3	7	5	10	9½	2½	....	12	5	2½
2½	7½	1	4	5⅞	⅞	¾	5½	3¾	7½	5½	11	10½	2½	....	13	5½	2½
3	8¼	1⅛	8	6⅝	⅞	¾	6	3½	8	6	12	11½	3	....	14½	6	3
3½	9	1 3/16	8	7¼	⅞	¾	6½	4	9	6½	13	12½	3	....	15½	6½	3
4	10	1¼	8	7⅞	⅞	¾	7	4¼	10	7	14	13½	3	....	16½	7	3
4½	10½	1 5/16	8	8½	⅞	¾	7½	4½	11	7½	15	14½	3½	13⅝	18	7½	3½
5	11	1⅜	8	9¼	⅞	¾	8	5	12	8	16	16	3½	14	19½	8	3½
6	12½	1 7/16	12	10⅝	⅞	¾	9	5½	13	9	18	17½	4	14⅝	21½	9	4
7	14	1½	12	11⅞	1	⅞	9½	5¾	14½	9½	19	19	4½	15⅜	23½	9½	4½
8	15	1⅝	12	13	1	⅞	10½	6	16	10½	21	20½	5	16	25½	10½	5
9	16¾	1¾	12	14	1⅛	1	11	6½	18	11	22	22¾	5	20⅞	27¾	11	5
10	18¼	1⅞	16	15¾	1⅛	1	12	7	20	12	24	25¼	5½	21⅝	30¾	12	5½
12	20¾	2	16	17¾	1¼	1⅛	13½	8	22	13½	27	27¾	6	23	33¾	13½	6
14	23½	2 3/16	20	20¼	1⅜	1¼	15	9½	24	15	30	31¼	6½	24⅜	37¾	15	6½
15	25	2⅛	20	21½	1⅜	1¼	15½	10	26	15½	31	33¼	6½	25⅛	39¾	15½	6½
16	26	2¼	20	22½	1½	1⅜	16½	10½	28	16½	33	34½	7½	25¾	42	16½	7½
18	28½	2⅜	24	24¾	1½	1⅜	17¾	11	30	17¾	35½	37¾	8	27⅛	45¾	17¾	8
20	31	2½	24	27	1⅝	1½	19¼	11½	32	19¼	38½	40¾	8½	28⅝	49¼	19¼	8½
22	33	2⅝	28	29¼	1⅝	1½	20½	12	34	20½	41	43¾	9½	30	53¼	20½	9½
24	36	2¾	28	32	1¾	1⅝	22½	13	36	22½	45	47½	10	31½	57½	22½	10

Bolts to be $\frac{1}{8}$ inch smaller in diameter than the bolt holes.

Bolt holes should straddle the center lines.

7. Size of all fittings scheduled indicates the inside diameter of the ports. For the outside diameter of pipe use the corresponding size of the inside diameter fittings.

7. Size of all fittings schedules indicates the inside diameter of ports except for extra-heavy fittings 14 inches and larger, when the port diameter is $\frac{3}{4}$ inch smaller than nominal size.

8. The face-to-face dimension of a reducer, either straight or eccentric, shall be equal to the diameter of the larger flange.

8. The face-to-face dimension of reducers, either straight or eccentric, for all pressures, shall be the same face to face as given in the table of dimensions.

9. Square head bolts with hexagonal nuts are recommended.

10. Twin ells, double-branch ells, side-outlet ells, side-outlet tees and fourway tees, whether straight sizes or reducing, carry the same dimensions center to face and face to face as the regular ells and tees.

10. Twin elbows, whether straight or reducing, carry same dimensions, center to face and face to face, as regular straight-size ells and tees.

Side-outlet elbows and side-outlet tees, whether straight or reducing sizes, carry same dimensions center to face and face to face as regular tees having the same reductions.

11. Bull-head tees or tees increasing on outlet will have the same center-to-face and face-to-face dimensions as a straight fitting of the size of the outlet.

12. Up to and including the 4-inch size, center-to-face and face-to-face dimensions of reducing fittings will be the same as that of a straight fitting of the larger opening.

12. Tees and crosses 9-inch and down, reducing on the outlet or run and outlet, use the same dimensions as straight sizes of the larger port.

Sizes 10-inch and up, reducing on the outlet, are made in two lengths, depending on the size of the outlet as given in the table of dimensions.

If the outlet is larger than that given in table, use dimensions of straight sizes.

Laterals $3\frac{1}{2}$-inch and down, reducing on the branch, use the same dimensions as straight sizes of the larger port.

Sizes 4-inch and up, reducing on the branch, are made in two lengths, depending on the size of the branch as given in the table of dimensions.

If the outlet is larger than that given in the table, use dimensions of the straight sizes.

Y's are special and are made to suit conditions.

Double-sweep tees are not made reducing on the run.

Steel flanges, fittings and valves are recommended for superheated steam.

13. Pipe sizes 14-inch and over refer to outside diameter.

TABLE 112.

COMPARATIVE COST OF VARIOUS PIPE FLANGE FITTINGS, 12-INCH PIPE.

(Circular from the Crane Company.)

	Screwed.	Shrunk.	Lap Joint. Long Hub.	Lap Joint. Short Hub.	Lap Joint. No Hub.	Welded.	Rolled.	Single Riveted.
Cast iron	$ 7.40	$16.00	$18.00	......	......	......	$13.00	$21.00
Ferrosteel	8.70	18.40	20.00	......	......	......	16.00	23.40
Malleable iron	9.90	......	......	$22.00	......	......	18.00	......
Cast steel	22.40	28.40	34.00	......	$33.00	......	25.00	33.40
Weldless steel	26.40	32.40	38.00	......	37.00	$41.00	30.00	37.40

Any of the above screwed, shrunk, welded, rolled, or single-riveted flanges can be furnished with male or female face at $1.25 extra.

The screwed or welded flanges can be furnished with tongued or grooved face at $1.25 extra.

Any of the above screwed, shrunk, or single-riveted flanges can be furnished with calking recess at $1.25 extra.

TABLE 113.

LOSS OF HEAT FROM BARE STEAM PIPE.*

Still Air.

Authority of Test.	Descriptive References.	Size of Pipe.	Square Feet of Heating Surface.	Steam Pressure, Gauge.	Steam Temperature.	Temperature of Air.	Difference of Temperature.	Pounds of Steam Condensed per Sq. Ft. per Hour.	B.T.U. per Sq. Ft. per Hr. per Deg. Dif. in Temp.
Barrus	Power, Dec., 1901; Trans. A.S.M.E., vol. XXIII; Stevens Ind., Vol. XIX, p. 388.	2	63.57	82	325	56	268.6	0.915	3.01
Do		2	63.92	149	365	63.3	302.2	1.150	3.25
Do		10	98.33	149	365	73.6	291.7	1.085	3.18
Hudson Beare	....	3.53†	8.13	135	358	67	291	1.050	3.10
" 130 lbs."	Stevens Ind., Vol. XIX, p. 388.	2	50.66	128	354	80.1	274.6	0.994	3.13
Jacobus	Stevens Ind., Vol. XVIII.	2	7.63	53	301	71.2	229.6	0.707	2.78
Brill	Trans. A.S.M.E., Vol. XVI.	8	135.4	110	344	75.5	269	0.834	2.71

* C. P. Paulding, *Stevens Indicator*, Vol. XIX, p. 388. † Outside diameter.

371. Coverings. — Steam pipes, feed-water pipes, boiler steam drums, receivers, separators, etc., should be covered with heat-insulating material to reduce radiation losses to a minimum. For most practical purposes the loss of heat from a bare steam pipe or drum may be taken as 3 B.t.u. per square foot per hour per degree difference in temperature, Table 80. The actual loss depends upon the diameter of the pipe, on its position whether vertical or horizontal, the nature of the surface, and the velocity of the surrounding air currents. For a detailed analysis of these various influences, and interesting information on the transmission of heat, the reader is referred to Paulding's "Steam in Covered and Bare Pipes."

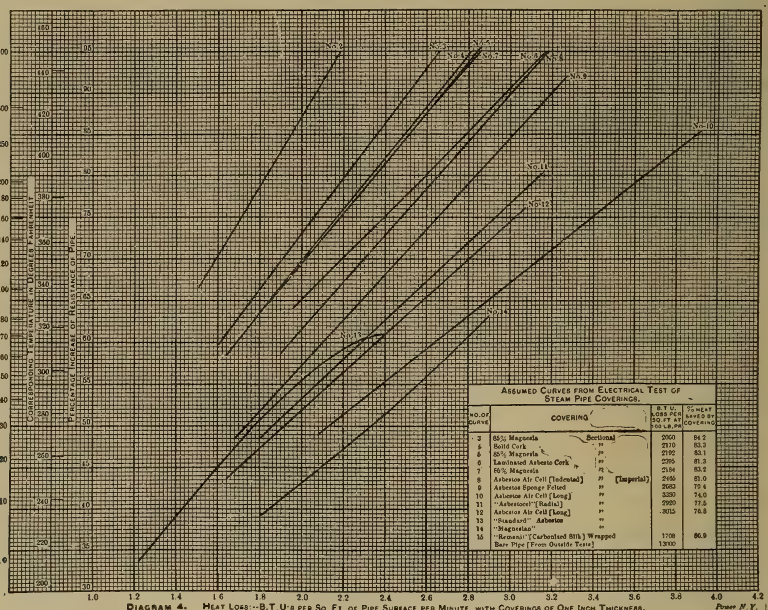


Fig. 462. Efficiency of Pipe Coverings.

By properly applying any good commercial covering, from 75 per cent to 90 per cent of the heat loss may be prevented. (See Fig. 462 and Table 114.)

Example: Required the saving per annum due to covering a pipe 10 inches in diameter and 100 feet long; steam pressure 150 pounds; average temperature of the air 76 degrees F.; cost of covering applied 65 cents per running foot; efficiency of covering 85 per cent; cost of coal $2.50 per ton; plant to operate 14 hours per day and 300 days per year.

The temperature of steam at 150 pounds pressure = 366 degrees F.

Difference of temperature between the steam and air = 366 − 76 = 290 degrees F.

Loss per square foot per hour, bare pipe = 3 × 290 = 870 B.t.u.

Loss per square foot per day, bare pipe = 870 × 14 = 12,180 B.t.u.

Loss per square foot per year, bare pipe = 12,180 × 300 = 3,654,000 B.t.u.

100 lineal feet of 10-inch pipe have an external surface of 282 square feet. Therefore the loss per year from the bare pipe is

$$282 \times 3,654,000 = 1,030,000,000 \text{ B.t.u. (approx.).}$$

TABLE 114.

EXPERIMENTS ON STEAM-PIPE COVERINGS.

("Condensation of Steam in Covered and Bare Pipes" [Paulding].)

Kind of Covering.	Diam. of Test Pipe, Inches.	Thickness of Covering, Inches.	Temperatures F.		B.T.U. per Hour per Square Foot of Pipe Surface.		Date of Test.	Testing Expert.
			Steam.	Air.	Total.	Per Degree Difference.		
Hair felt	2	0.96	302.8	71.4	89.6	0.387	1901	Jacobus
Do	8	0.82	348.3	69.0	117.9	0.422	1894	Brill
Remanit for intermediate pressure.	2	0.88	304.5	73.3	100.3	0.434	1901	Jacobus
Remanit for high pressure.	2	1.30	306.6	76.1	83.7	0.363	1901	Jacobus
Mineral wool	8	1.30	344.1	58.3	81.3	0.284	1894	Brill
Champion mineral wool	8	1.44	346.1	74.3	86.1	0.317	1894	Brill
Rock wool	8	1.60	344.1	63.0	72.0	0.256	1894	Brill
Asbestos sponge felted	2	1.125	364.8	60.7	145.0	0.477	1901	Barrus
Do	10	1.375	364.8	62.8	85.0	0.248	1901	Barrus
Do	2	1.14	309.2	79.4	59.7	0.260	1901	Jacobus
Magnesia	4	1.12	388.0	72.0	147.0	0.465	1896	Norton
Do	2	1.09	354.7	80.1	155.8	0.567	1896	Paulding
Do	8	1.25	344.1	66.3	106.6	0.384	1895	Brill
Do	2	1.08	310.9	81.6	69.8	0.304	1901	Jacobus
Do	2	1.00	365.2	64.6	155.0	0.515	1901	Barrus
Do	10	1.19	365.2	66.0	103.0	0.347	1901	Barrus
Asbestos, Navy Brand	2	1.20	309.2	79.4	69.9	0.304	1901	Jacobus
Do	2	1.125	365.2	64.6	176.0	0.585	1901	Barrus
Do	10	1.375	365.2	66.8	112.0	0.375	1901	Barrus
Manville sectional	8	1.70	345.5	78.3	93.4	0.394	1894	Brill
Do	2	1.31	354.7	80.1	157.0	0.572	1896	Paulding
Do	4	1.25	388.0	72.0	143.0	0.453	1896	Norton
Asbestos air cell	4	1.12	388.0	72.0	166.0	0.525	1896	Norton
Do	2	0.96	303.3	72.3	165.5	0.716	1901	Jacobus
Asbestos fire felt	8	1.30	344.7	79.0	133.5	0.502	1894	Brill
Do	2	1.00	354.7	80.1	198 0	0.721	1896	Paulding
Do	2	0.99	307.4	72.5	180.0	0.766	1901	Jacobus
Fossil meal	8	0.75	347.1	75.3	238.0	0.876	1894	Brill
Riley cement	8	0.75	347.9	74.3	260.0	0.950	1894	Brill

Assuming a net available heat value of 10,000 B.t.u. per pound for the coal, the equivalent coal consumption is 51.5 tons, valued at 51.5 $\times$ \$2.50 = \$128.75.

The covering will save 85 per cent of this, or \$109.50 per annum.

The pipe covering applied will cost 100 $\times$ \$0.65 = \$65.00.

In this case the covering will pay for itself in considerably less than a year.

Pipe covering is applied in sections molded to the required form and held to the pipe by bands, or may be applied in a plastic form. The former is more readily applied and removed, and is usually adopted for pipes, while the valves and fittings are sometimes covered with plastic material. Piping should be tested under pressure before being covered, since leaks destroy the efficiency and life of the covering. If the surrounding atmosphere is moist the covering should be given two or three coats of good paint. Coverings are sometimes applied to cold-water pipe to prevent sweating in a humid atmosphere.

Identification of Power House Piping by Colors: Power and Engr., Apr. 26, 1910, p. 752.

372. Expansion. — One of the most difficult problems in the design of a piping system is the proper provision for expansion and contraction due to change in temperature. If a pipe is immovably fixed at both ends and under no strain when cold, and the temperature is increased, as by the admission of steam, it is subjected to a compression proportional to the rise in temperature (within the elastic limit). For example, a 6-inch standard extra-heavy wrought-iron pipe 200 feet long at 66 degrees F., if heated to 366 degrees F. (the temperature corresponding to steam at 165 pounds per square inch absolute pressure), will exert an axial force of

$$P = EA\,(t_1 - t)\,\mu. \quad \text{(Mechanics of Engng., Church, p. 218.)} \qquad (254)$$

P = force in pounds.
E = modulus of elasticity, 30,000,000.
t_1 = final temperature, degrees F.
t = initial temperature.
μ = coefficient of expansion, 0.0000075.
A = sectional area of the pipe material, 8.5 square inches.

Hence

$$P = 30{,}000{,}000 \times 8.5\,(366 - 66)\,0.0000075$$
$$= 573{,}750 \text{ pounds.}$$

Unless well braced throughout its entire length the pipe will buckle and become distorted. If free to expand its length would increase.

The temperature of the pipe is always less than that of the steam on account of radiation from the outer surface and varies with the efficiency of the covering. But ignoring radiation the increase in length due to temperature increase is

$$l = \mu\,(t_1 - t)\,L, \tag{255}$$

in which

l = increase in length, inches.
L = length of pipe, inches.
Other notations as in (254).

Substituting in (254), $t_1 = 366.$
$t = 66.$
$\mu = 0.0000075.$
$L = 2400.$
$l = 0.0000075\,(366 - 66)\,2400$
$= 5.4$ inches.

The total increase in length will be the sum of the elongation due to pressure and that due to increase in temperature.

Since the forces produced by expansion are practically irresistible, the pipe is invariably allowed to expand freely by suitable means so as not to strain the connections. The coefficients of expansion per degree difference in temperature for various pipe materials are given in Table 115.

Headers less than 50 feet in length usually require no special provisions for expansion, provided the ends are free and the leads to and

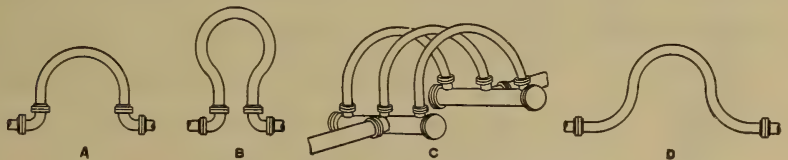


FIG. 463. Types of Expansion Pipe Bends.

from the header are not too short, the pipe usually being anchored at the middle and permitted to expand in either direction. Free expansion of the feeders may be provided for

1. By *long radius bends,* as in Fig. 463.
2. By *double-swing screwed fittings,* as in Fig. 464, or
3. By *packed expansion joints,* Fig. 465.

Where practicable the long radius bends will prove most satisfactory. The radius of the bend should not be less than 5 diameters of the pipe, and larger if possible. The length of straight pipe at the end of each bend should not be less than twice the diameter of the pipe measured from the face of the flange.

On account of the great strains to which the joints of pipe bends are subjected, the welded joint, *G*, Fig. 461, is recommended as giving the best results. The next best is the lap joint, *F*, Fig. 461.

TABLE 115.

COEFFICIENTS OF LINEAR EXPANSION PIPING MATERIALS.

Material.	Temperature Range.	Mean Coefficient per Degree F.
Wrought iron and mild steel	32–212	0.00000656
Wrought iron	32–572	0.00000895
Cast iron	32–212	0.00000618
Cast steel	32–212	0.00000600
Hardened steel	32–212	0.00000689
Nickel-steel, 36 per cent Nickel	32–572	0.00000030
Copper, cast	32–212	0.00000955
Copper, wrought	32–572	0.00001092
Lead	32–212	0.00001580
Cast brass	32–212	0.00001043
Brass wire and sheets	32–212	0.00001075
Tin cast	32–212	0.00001207
Tin hammered	32–212	0.00001500
Zinc cast	32–212	0.00001633
Zinc hammered	32–212	0.00001722

LINEAR EXPANSION OR CONTRACTION OF CAST IRON IN INCHES PER 100 FEET, — DEGREES F.

Temperature Difference.	Expansion.	Temperature Difference.	Expansion.
100	0.72	300	2.376
150	1.1016	400	3.360
200	1.5024	500	4.440
250	1.9260	600	5.616
....	....	800	7.872

Multiply by 1.1 for wrought mild steel.
Multiply by 1.5 for wrought copper.
Multiply by 1.6 for wrought brass.

Fig. 463, *A*, *B*, *C*, *D*, shows applications of pipe bends to straight pipe runs. *A* is the cheapest and most common arrangement for all sizes of pipe. *B* is a modification for limited center-to-center spaces. *C* shows a common method of taking up expansion in straight runs of pipe of very large diameters where the space requirements prohibit the use of a single U bend. Here the main runs are connected to manifolds which in turn are connected by a number of small U bends, the equivalent areas of which correspond to that of the large pipes. This makes a

more flexible connection than if a single U bend were used. The arrangement D does away with the elbows required in A, but is not applicable to pipes over 8 inches in diameter.

Figs. 478 and 479 show applications of pipe bends to boiler and header connections.

Fig. 464 shows a double-swing screwed joint in which expansion causes the fittings to turn slightly and thus relieve the strain. This method is usually adopted where long radius bends are not practicable on account of lack of space and where screwed fittings are used.

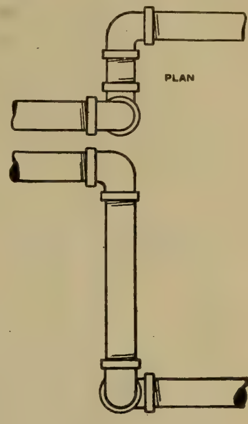


FIG. 464. "Double-swing" Expansion Joint.

Slip joints, Fig. 465, are now little used except with very large pipes and where space prohibits long radius bends. When slip joints are employed the pipe must be securely anchored to prevent the steam pressure from forcing the joint apart and at the same time permit the pipe in expanding to work freely in the stuffing box. Sagging of the pipe on either side, which might cause binding in the joint, is prevented by suitable supports.

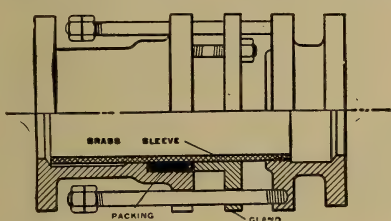


FIG. 465. Slip Expansion Joint.

Expansion in Steam Pipes: Power, July, 1906, p. 426, Jan., 1904, p. 30, March, 1904, p. 160, Oct., 1904, p. 609, Dec., 1900; Am. Elecn., 10–432; Engr., U. S., Feb. 1, 1904, p. 125; Eng. News, 44–194, 47–468, 50–487; Power, June 2, 1908.

373. Pipe Supports and Anchors. — Pipe lines must be supported to guard against excessive deflection and vibration. Supports are conveniently classified as (1) hangers, (2) wall brackets, and (3) floor stands.

Fig. 466 illustrates a type of hanger for suspending pipes from I beams. The supports being free to swing, no provision for expansion is necessary. A properly designed hanger may be readily removed without disturbing the pipe line, and should be adjustable to facilitate "lining up." If of rigid construction the lower end should be provided with a roller.

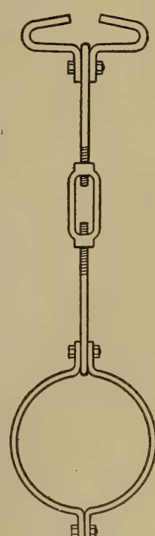


FIG. 466. A Typical Pipe Hanger.

Fig. 467 gives the details of a wall bracket with rolls and roll binder. Supports adjacent to long radius bends should be provided with roll binders as illustrated to prevent the pipe

from springing laterally, but they may otherwise be omitted. The rollers are often made adjustable to facilitate lining up.

Fig. 468 illustrates a typical floor stand. Pipe lines are usually securely anchored at suitable points in a manner similar to that illustrated in Fig. 469, the pipe resting on a saddle and being rigidly clamped to the bracket by a flat iron band with ends threaded and bolted. This limits expansion to one direction and prevents excessive strain on the fittings.

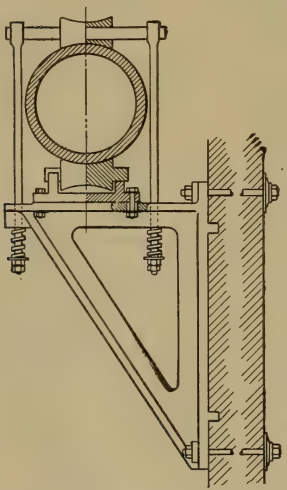


Fig. 467. A Typical Wall Bracket with Binding Roll.

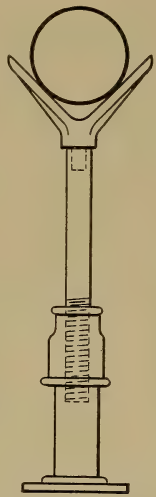


Fig. 468. A Typical Floor Stand.

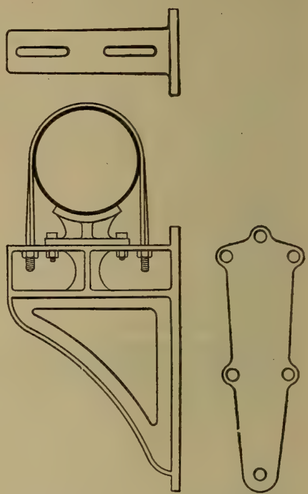


Fig. 469. A Typical Pipe Anchor.

Fig. 470 illustrates a method of suspending and counterbalancing expansion loops in a main header and Fig. 471 a flexible support for a large vertical exhaust header.

374. General Arrangement of High-pressure Steam Piping. — The general arrangement of piping depends in a great measure upon the space available for engines and boilers.

The engine and boiler room may be placed

(1) Back to back, Fig. 473.
(2) End to end, Fig. 472.
(3) Double decked, Fig. 480.

The *back-to-back* arrangement is the most common and, other things permitting, is to be preferred on account of the short and direct connection between engines and boilers and the ease of enlargement. The engine and boiler rooms are separated by a wall, and as much of the piping as possible is located in the boiler room.

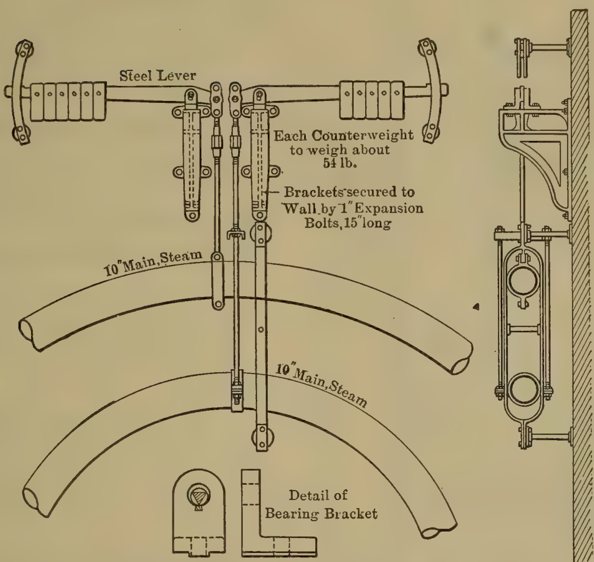


FIG. 470. Method of Suspending and Counterbalancing Expansion Loops in Steam Mains.

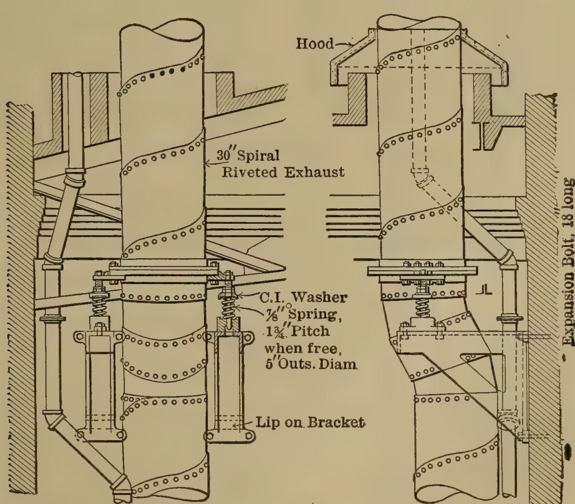


FIG. 471. Spring Support for 30-inch Exhaust Pipe.

The *end-to-end* arrangement is ordinarily limited to situations where the distribution of space precludes the back-to-back system.

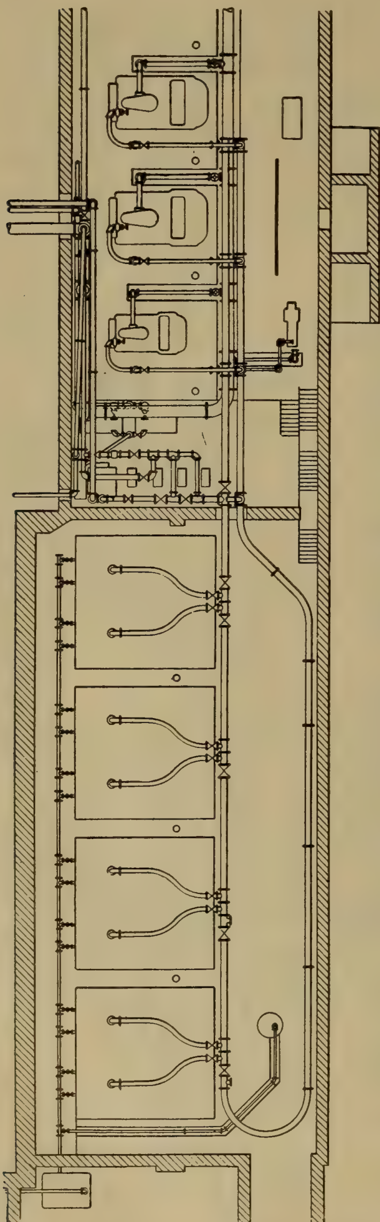


Fig. 472. Plan of High-pressure Piping, Princeton University Power Plant.

The *double-decked* arrangement is frequently used where ground space is limited or expensive.

Engines and boilers are connected in a variety of ways through steam headers as shown in the following examples:

1. Spider system, Fig. 473.
2. Single header, Fig. 474.
3. Duplicate headers, Figs. 475 and 476.
4. Loop or ring header, Fig. 477.
5. The "unit" system, Fig. 478.

The *spider* system is often used in small plants. In this arrangement all branch pipes are brought to one central header which is made as short as possible. The shortness of such a header minimizes danger from breakdowns, and brings all the principal valves close together.

The *single-header* system is perhaps the most common, since it embodies simplicity, low first cost, and provision for extension.

The *duplicate* system is losing favor, since experience shows that the extra cost of the duplicate mains will usually give better returns in continuity of operation and maintenance if invested in high-grade fittings on a single-pipe system. A small auxiliary header is used in modern plants where double mains are required; see Fig. 476.

The *loop header* is well adapted for the power plants of tall office buildings, Fig. 481, in which a large number of steam engines, elevator pumps, air compressors, and miscellaneous steam-consuming appliances are crowded together in a comparatively small space.

Large modern power plants are, by the latest practice, divided into complete and independent units, as in Fig. 478, each prime mover having its own boiler equipment, coal and ash-handling machinery, feed pumps, and piping, operated independently of the rest of the plant, though provision is made whereby any boiler equipment may provide steam for any prime mover.

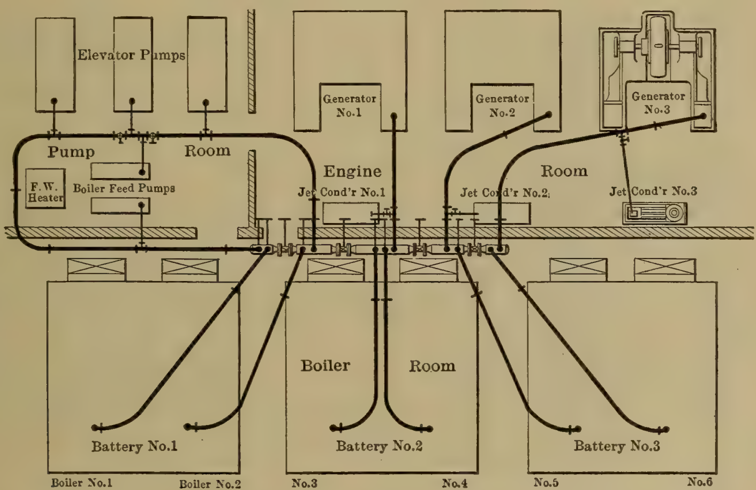


FIG. 473. "Spider" System.

The power plant of the Manhattan Elevated Railway Company, New York, is practically divided into eight sections each consisting of an engine and eight boilers, the boilers being "double decked" (Fig. 480).

The branch pipes from the upper and lower batteries lead into 18-inch headers, the steam from each being conducted to a receiver reservoir 36 inches in diameter and 20 feet long in the engine-room basement directly behind each engine, from which the two high-pressure cylinders are supplied. Gate valves are used in each boiler branch, one close to the boiler and another near the header, and also in the steam pipes near the reservoir. The steam headers for each of the eight units are connected by a main which equalizes the pressure and allows a deficiency in one unit to be made up from the others.

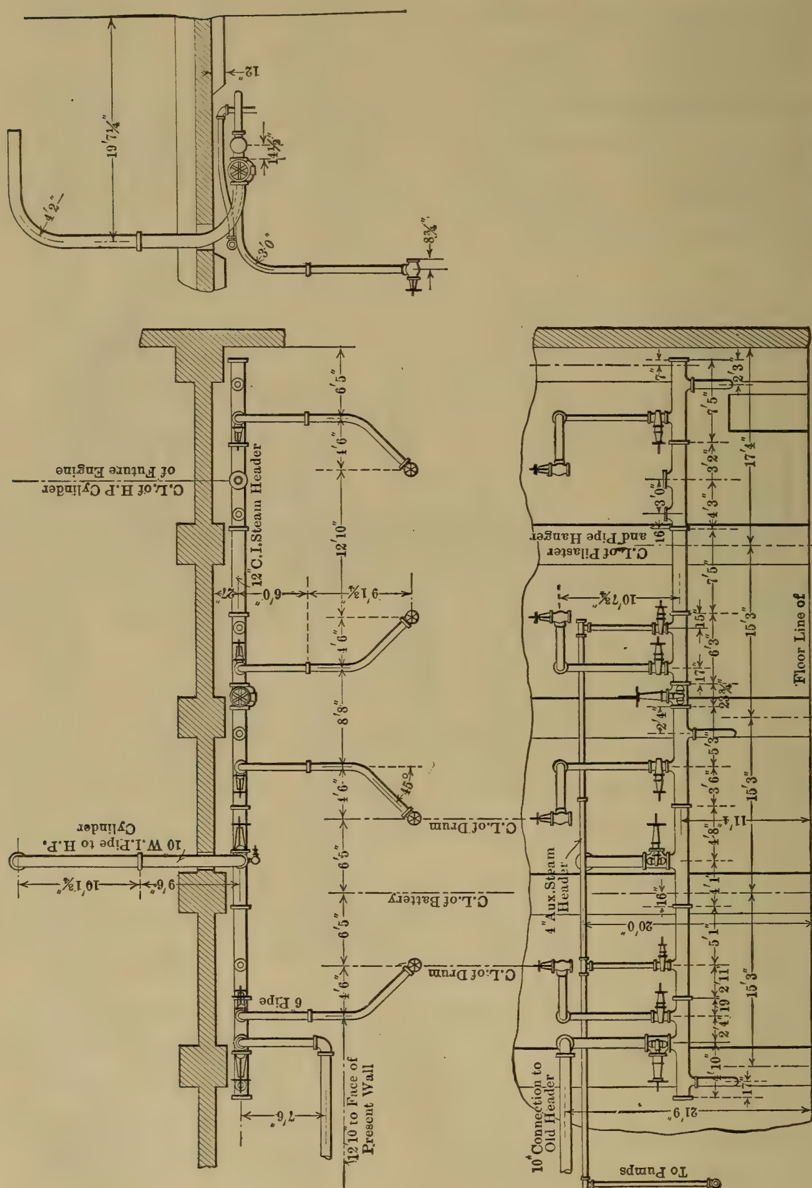


FIG. 474. Steam Header and Branches, Des Moines City Ry. Co.'s Power House. (Typical Single Header.)

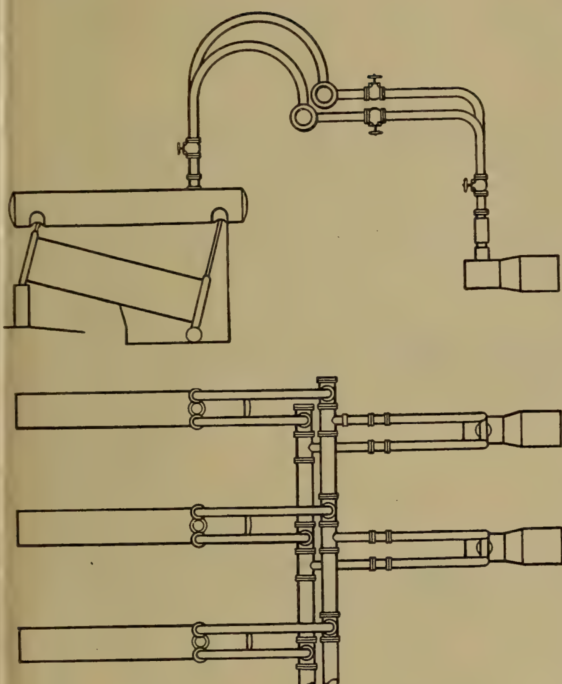


FIG. 475. Typical Duplicate Header System.

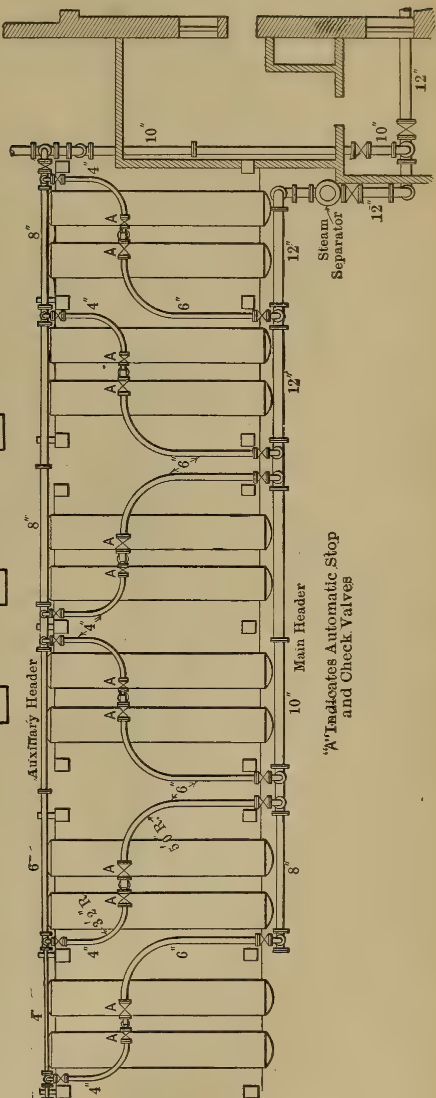


FIG. 476. Typical Auxiliary Header System.

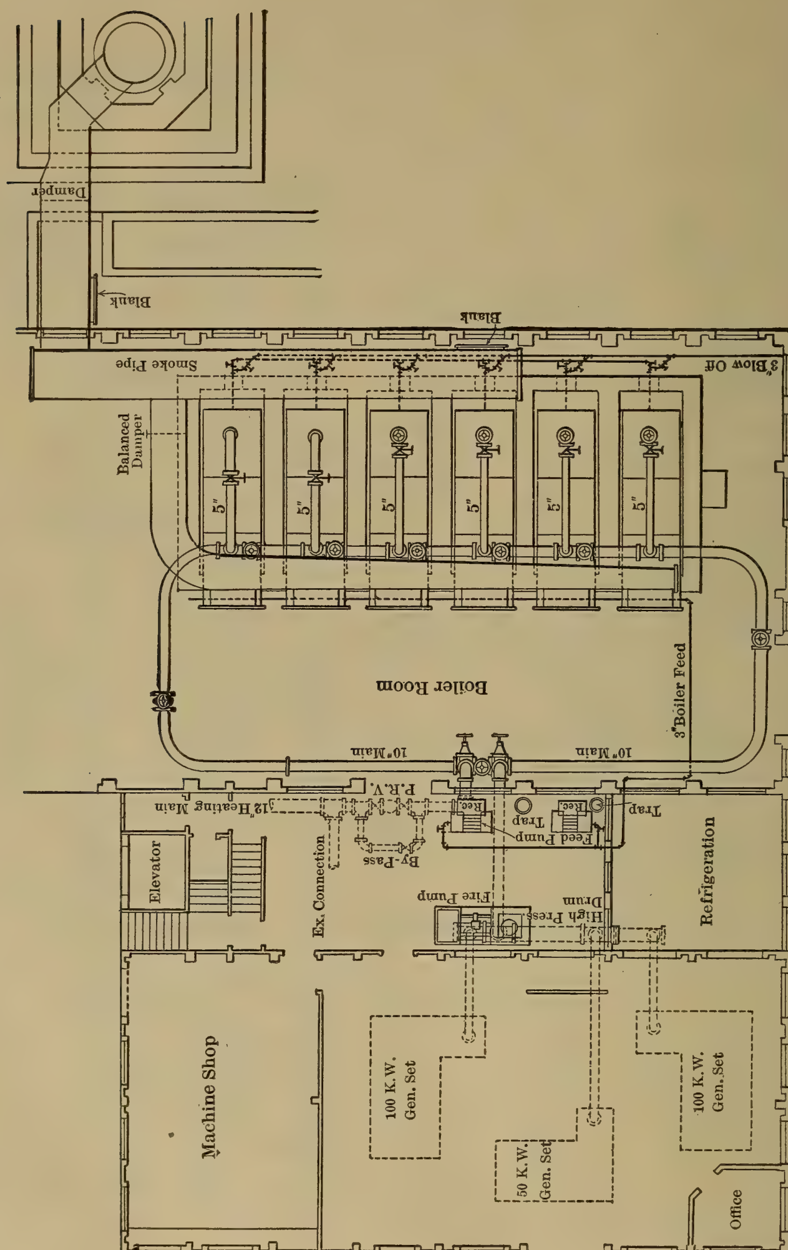


FIG. 477. Typical Loop Header.

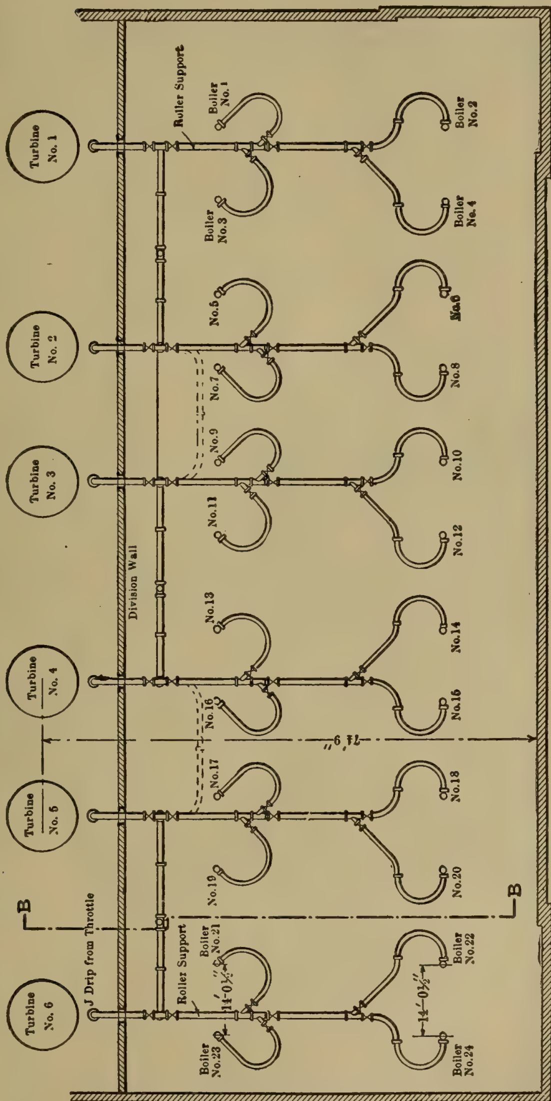


FIG. 478. Plan of High-pressure Piping, Yonkers Power House of the New York Central R.R.

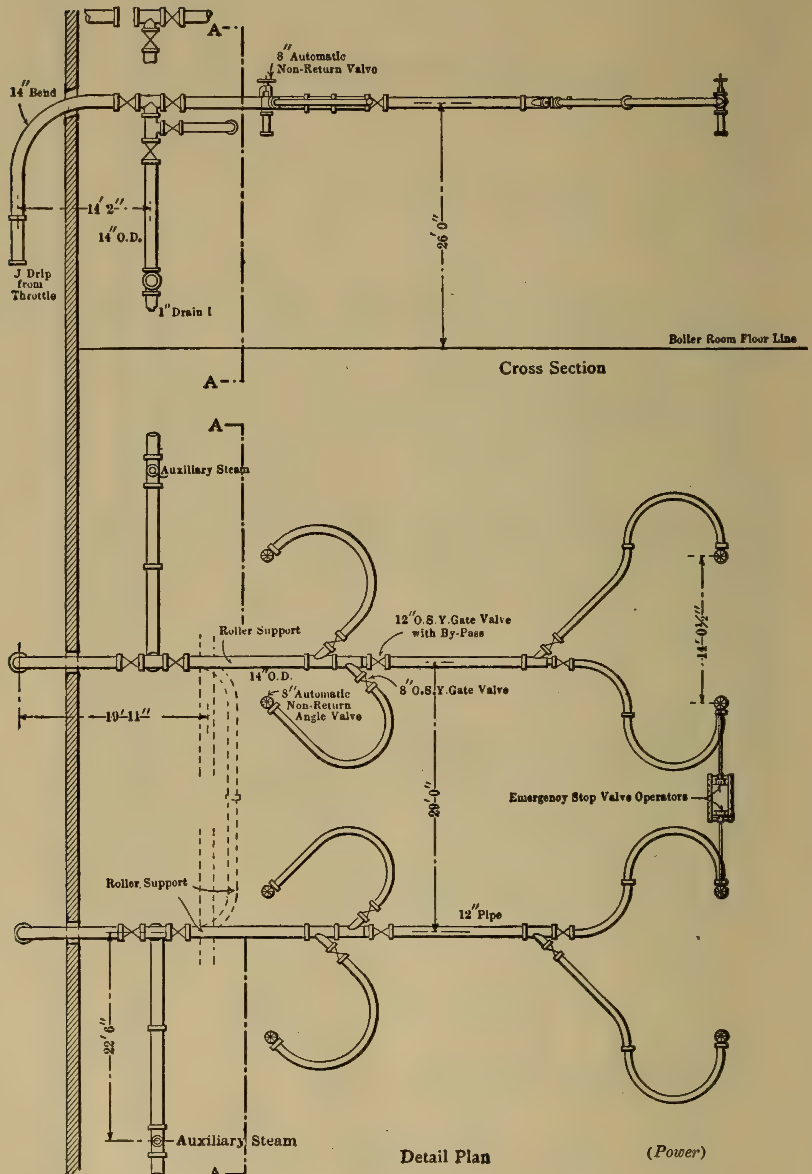


FIG. 479. Details of Boiler Steam Piping, Yonkers Power House of the New York Central R.R.

Figs. 478 and 479 show the general arrangement of the steam piping at the Yonkers power house of the New York Central. The turbines are connected in pairs by 14-inch loops, each turbine taking steam from either of two banks of four boilers. The high-pressure steam piping is of mild steel with modified reënforced "Van Stone" joints. The high-pressure valves are of the split-disk pattern with semi-steel bodies. Expansion is taken up by the long sweep bends.

Plants using superheated steam are ordinarily piped to supply saturated steam to the auxiliaries as illustrated in Fig. 482. The boiler branch E, leading to the main header, normally supplies superheated steam to the engines. C is an auxiliary main supplying the air pumps, stoker engines. and other auxiliaries with saturated steam from branch pipe D.

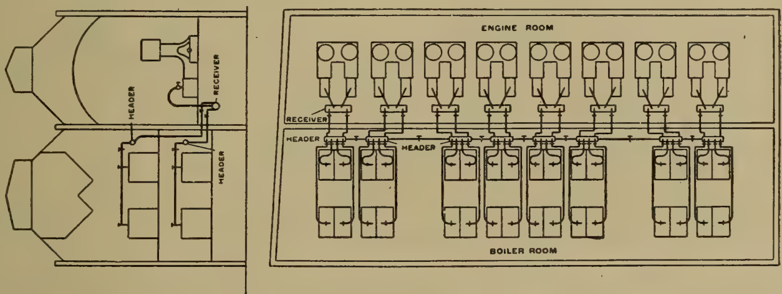


Fig. 480. General Arrangement of Steam Piping, Manhattan Elevated Station.

375. Main Steam Headers. — Until quite recently it was the usual practice to make the area of the steam header equivalent to the combined areas of the feeders, but the function of the header is now regarded as that of an equalizer rather than a storage reservoir. In the various large power houses recently built in New York City, with ultimate capacities of from 100,000 to 250,000 kilowatts, the largest steam headers are not over 16 inches in diameter. In some recent designs the pipes leading from the header to the engines are two sizes smaller than called for by the engine builders. In this case large receiver separators two to four times the volume of the high-pressure cylinder are provided near the throttle as in Fig. 480. The pipes between receiver and engine are full size. The object of the arrangement is to give (1) a constant flow of steam, (2) a full supply of steam close to the throttle, and (3) a cushion near the engine for absorbing the shock caused by cut-off. With moderately superheated steam and boiler pressures from 125 to 150 pounds a velocity of 8000 feet per minute is allowed in the header and as high as 9000 feet per minute between header and receiver. With steam turbines velocities as high as 12,000 feet per minute are per-

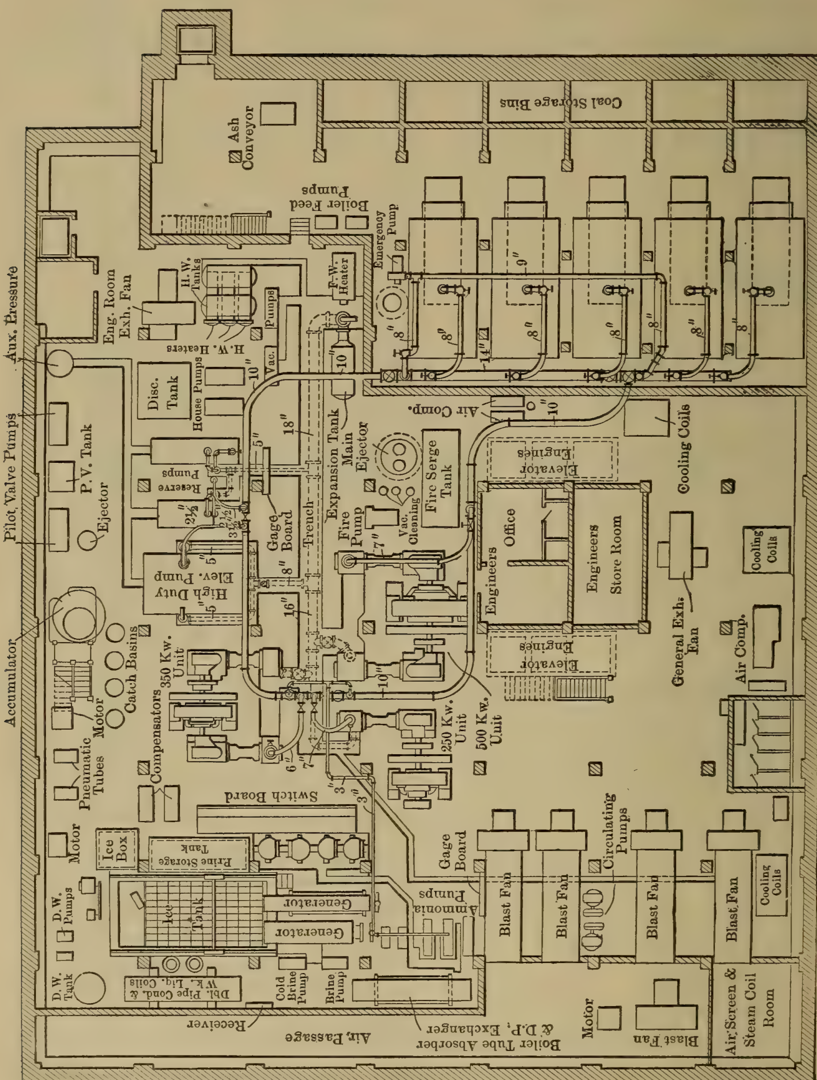


FIG. 481. General Arrangement of Steam and Exhaust Piping, La Salle Hotel, Chicago, Ill.

missible, provided the pipe is less than 50 feet in length and practically free from sharp bends. Main headers are ordinarily constructed of mild steel, though cast-iron and cast-steel headers are not uncommon.

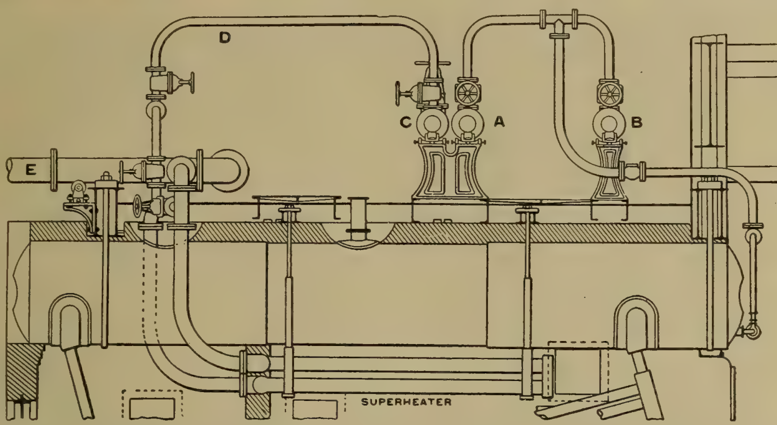


Fig. 482. Overhead Boiler Piping, Quincy Point Power Plant of the Old Colony St. Ry. Co., Quincy Point, Mass.

Cast headers permit of fewer joints and are well adapted to situations where a number of branches are closely grouped as in Fig. 473.

The proper arrangement and number of valves in the main header and feeders has been a subject of much consideration. Figs. 472 to 483

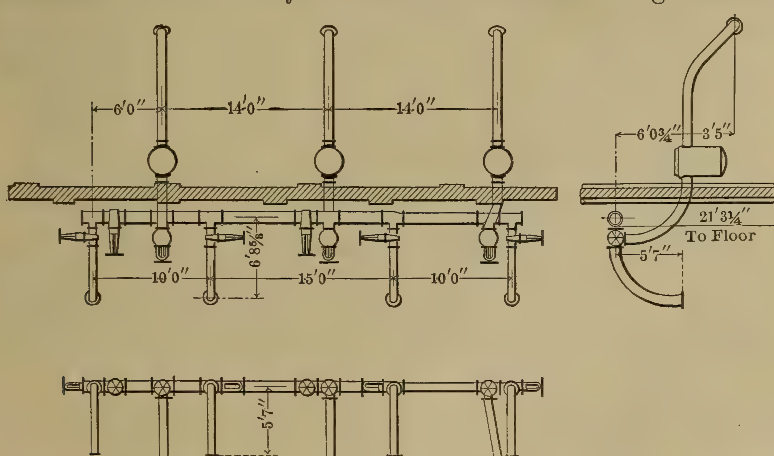


Fig. 483. Steam Header and Branches, Grand Rapids, Grand Haven and Muskegon Ry. Co. Power House.

show some of the different successful arrangements in recent installations. Where two valves are placed in a feeder they should be arranged so as not to form a pocket for the accumulation of leakage. In a number

of recent installations, Fig. 479, the valve nearest the boiler is of the "automatic stop and check" type, its function being the automatic cutting off of the steam from the header should the pressure in the boiler suddenly drop as in case of blowing out a tube.

Arrangement of Steam Piping: Power and Engr., Jan. 18, 1910, p. 117, Sept. 29, 1908, p. 523, Feb. 22, 1908; Engr. U. S., Dec. 1, 1904; Mech. Engr., Nov. 4, 1905; Power, Feb. 23, 1909, p. 363; Eng. News, Nov. 26, 1903, p. 487; Elec. Rev., Lond., Aug. 11, 1899, p. 251; St. Ry. Rev., Jan., 1900, p. 12; Nov., 1904, p. 869.

376. Flow of Steam in Pipes.* — The several accepted formulas relating to the flow of steam in pipes have been based upon a few experiments limited to pipes of small diameter; hence the application of these formulas to larger pipes or to conditions other than those under which they were deduced is apt to lead to considerable error. In small plants extreme accuracy in determining the proper sizes is not necessary; it is better to err in the installation of too large a pipe than one too small. In larger stations, however, where the pipes are large and the pressure is high, the cost of the piping increases very rapidly with the size. For example, the cost of 10-inch high-pressure fittings is from 15 to 20 per cent greater than 9-inch fittings, and in large installations this first cost item may be of considerable importance.

The simplest and most commonly used formula is based upon an allowable steam velocity of 6000 feet per minute, friction and other causes of drop in pressure being disregarded; thus, for a velocity of 6000 feet per minute,

$$d = 0.175\left(\frac{W}{\gamma}\right)^{\frac{1}{2}}, \tag{256}$$

in which

d = diameter of the pipe in inches.

γ = density of the steam in pounds per cubic foot, and

W = weight of steam flowing in pounds per minute.

In determining the diameter of the steam pipe opening for reciprocating engines a much lower velocity than 6000 feet per minute is assumed, to allow for the various conditions of operation. Average practice gives the constant in equation (256) a value of 0.3 instead of 0.175 when used in this connection.

Equation (256) gives satisfactory results for pipes under 100 feet in length and between 4 and 8 inches in diameter; for larger diameters the velocity could be increased with advantage; for smaller diameters or greater lengths friction and condensation would cause considerable drop in pressure and some one of the approved formulas in Table 116 should be used instead.

* See author's paper, Power, June, 1907, p. 377.

A large drop in pressure means a small pipe and high velocity with consequent decrease in condensation, but a point is soon reached where the economy in the size of pipe is more than offset by the loss

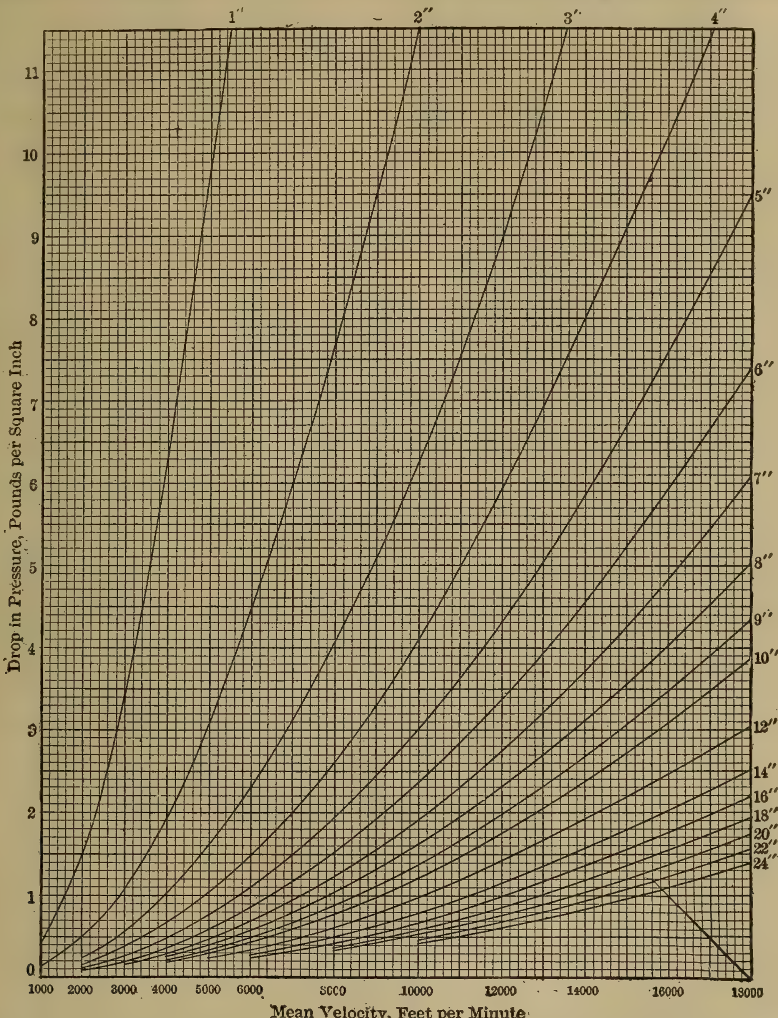


FIG. 484. Drop in Pressure for Various Velocities and Pipe Sizes. Initial Pressure 100 Pounds Gauge, Length of Pipe 100 feet.

in friction. There seems to be no fixed rule for determining the drop most suitable for any given set of conditions. In current practice the drop in pressure between boiler and engine ranges from a fraction of one pound to four pounds per square inch per 100 feet of pipe, with an average between one and two pounds.

TABLE 116.

FORMULAS FOR THE FLOW OF STEAM IN PIPES.

No. of Equation.	Author.	Reference.	Formula as Given by the Author.	Formula Reduced to Equivalent Notations. English Units. *
11	G. H. Babcock	"Steam" (1890). Pub. by B. & W. Co.	$W = 87 \left\{ \frac{yPd^5}{L\left(1 + \frac{3.6}{d}\right)} \right\}^{\frac{1}{2}}$	$W = 87 \left\{ \frac{yPd^5}{L\left(1 + \frac{3.6}{d}\right)} \right\}^{\frac{1}{2}}$
12	R. C. Carpenter	Trans. A.S.M.E., Vol. XX, p. 347.	$P = 0.0001306 \frac{W^2 L\left(1 + \frac{3.6}{d}\right)}{yd^5}$	$P = 0.0001306 \frac{W^2 L\left(1 + \frac{3.6}{d}\right)}{yd^5}$
7	Gutermuth	Zeit. d. Ver. D. Ing., April 16, 1904, p. 562.	$P = 0.0015\, y \frac{L}{d} V^2$	$P = 105.8 \cdot 10^{-10} y \frac{L}{d} V$
9	Geipel and Kilgour	London Electrician, May 5, 1893.	$P = \frac{W^2 L}{2525\, yd^5}$	$P = \frac{W^2 L}{2525\, yd^5}$
5	Hawksley	Pro. Inst. C. E., Vol. XXXIII, p. 55.	$V = 48 \left\{ \frac{Hd}{L} \right\}^{\frac{1}{2}}$	$V = 9976 \left\{ \frac{Pd}{yL} \right\}^{\frac{1}{2}}$
15	Hurst	King's "Treatise on Coal Gas," Vol. II, p. 374.	$Q = 1350 \left\{ \frac{D^5 P}{GL} \right\}^{\frac{1}{2}}$	$V = 10{,}360 \left\{ \frac{Pd}{yL} \right\}^{\frac{1}{2}}$
14	Ledoux	Annales des Mines, Vol. II, 1892.	$d = 0.056 \left\{ \frac{W^2 L}{P_0^{1.94} - P_1^{1.94}} \right\}^{\frac{1}{5}}$	$d = 0.699 \left\{ \frac{W^2 L}{P_1^{1.94} - P_2^{1.94}} \right\}^{\frac{1}{5}}$
9a	Martin	Engineering, March 19, 1897	$P = \frac{W^2 L}{3192\, yd^5}$	$P = \frac{W^2 L}{3192\, yd^5}$
4	Unwin	"Encyclopedia Britannica," Vol. XII, pp. 508, 516.	$H = 0.0027\left(1 + \frac{3}{10\,D}\right)\frac{V^2}{2\,g} \cdot \frac{4\,L}{D}$	$P = 0.0001306 \frac{W^2 L\left(1 + \frac{3.6}{d}\right)}{yd^5}$

* All notations as follows:
P_1 = Initial pressure, pounds per square inch absolute.
P_2 = Final pressure, pounds per square inch absolute.
$P = P_1 - P_2$ = Drop in pressure.
W = Weight of steam flowing, pounds per minute.
L = Length of pipe, feet.
d = Diameter of pipe, inches.
y = Mean density, pounds per cubic foot.

TABLE 117.

COMPARISON OF PIPE FORMULAS FOR THE FLOW OF STEAM.

	Author.	V = Velocity, Feet per Minute.	W = Weight, Pounds per Minute.	P = Drop in Pressure, Pounds per Square Inch.	d = Diameter, Inches.
Group II.	Geipel and Kilgour	$V = 9240\sqrt{\frac{Pd}{yL}}$	$W = 50.2\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003960\frac{W^2L}{yd^5}$	$d = 0.2087\sqrt[5]{\frac{W^2L}{Py}}$
	Gutermuth . .	$V = 9722\sqrt{\frac{Pd}{yL}}$	$W = 53\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003557\frac{W^2L}{yd^5}$	$d = 0.2032\sqrt[5]{\frac{W^2L}{Py}}$
	Hawksley . . .	$V = 9976\sqrt{\frac{Pd}{yL}}$	$W = 54.4\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003370\frac{W^2L}{yd^5}$	$d = 0.2010\sqrt[5]{\frac{W^2L}{Py}}$
	Martin.	$V = 10{,}350\sqrt{\frac{Pd}{yL}}$	$W = 56.5\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003133\frac{W^2L}{yd^5}$	$d = 0.1990\sqrt[5]{\frac{W^2L}{Py}}$
	Hurst.	$V = 10{,}360\sqrt{\frac{Pd}{yL}}$	$W = 56.5\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003126\frac{W^2L}{yd^5}$	$d = 0.1990\sqrt[5]{\frac{W^2L}{Py}}$
Group I.	Babcock	$V = 15{,}950\sqrt{\frac{Pd}{yL\left(1+\frac{3.6}{d}\right)}}$	$W = 87\sqrt{\frac{Pyd^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001321\frac{W^2L\left(1+\frac{3.6}{d}\right)}{yd^5}$	. . . .
	Unwin.	$V = 16{,}050\sqrt{\frac{Pd}{yL\left(1+\frac{3.6}{d}\right)}}$	$W = 87.5\sqrt{\frac{Pyd^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001306\frac{W^2L\left(1+\frac{3.6}{d}\right)}{yd^5}$	. . . .
	Carpenter . . .	$V = 16{,}050\sqrt{\frac{Pd}{yL\left(1+\frac{3.6}{d}\right)}}$	$W = 87.5\sqrt{\frac{Pyd^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001306\frac{W^2L\left(1+\frac{3.6}{d}\right)}{yd^5}$	. . . .
	Ledoux.	$V = 442\sqrt{\frac{(P_1^{1.94} - P_2^{1.94})\,d}{y^2L}}$	$W = 2.44\sqrt{\frac{(P_1^{1.94} - P_2^{1.94})d^5}{L}}$	$P_1^{1.94} - P_2^{1.94} = 0.1669\frac{W^2L}{d^5}$	$d = 0.699\sqrt[5]{\frac{W^2L}{P_1^{1.94} - P_2^{1.94}}}$

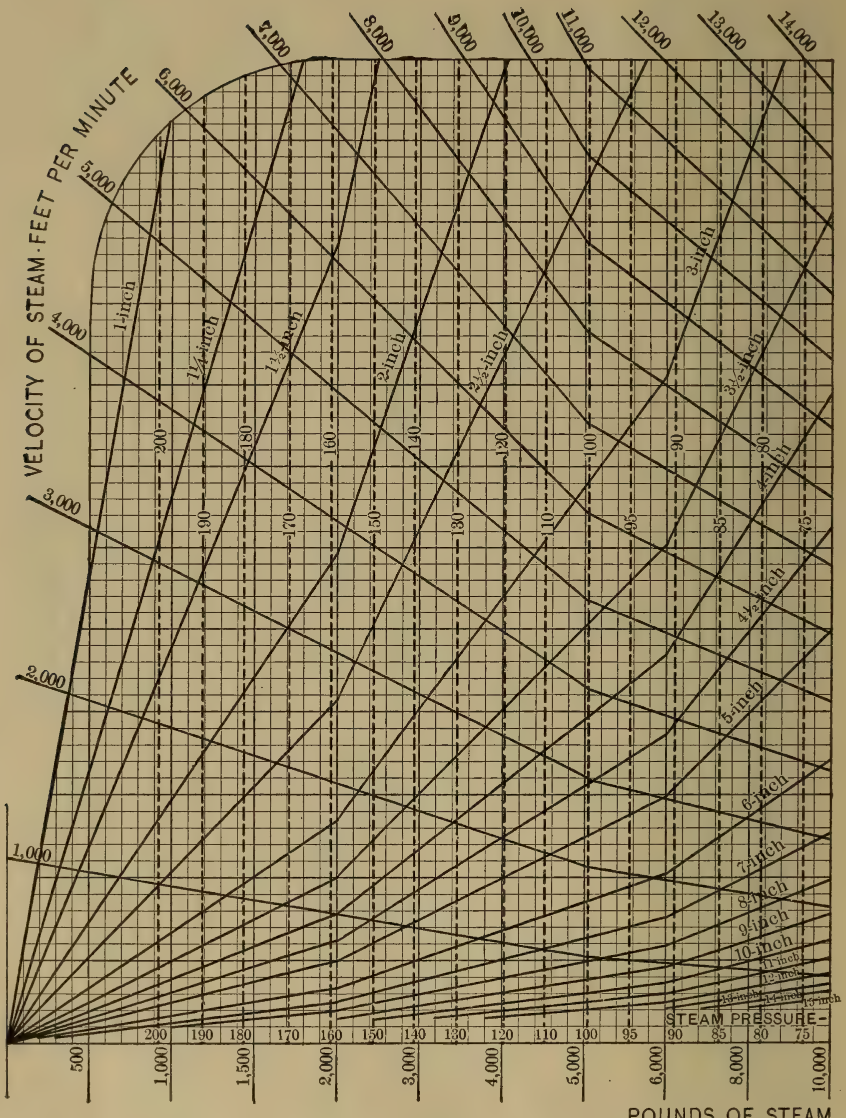
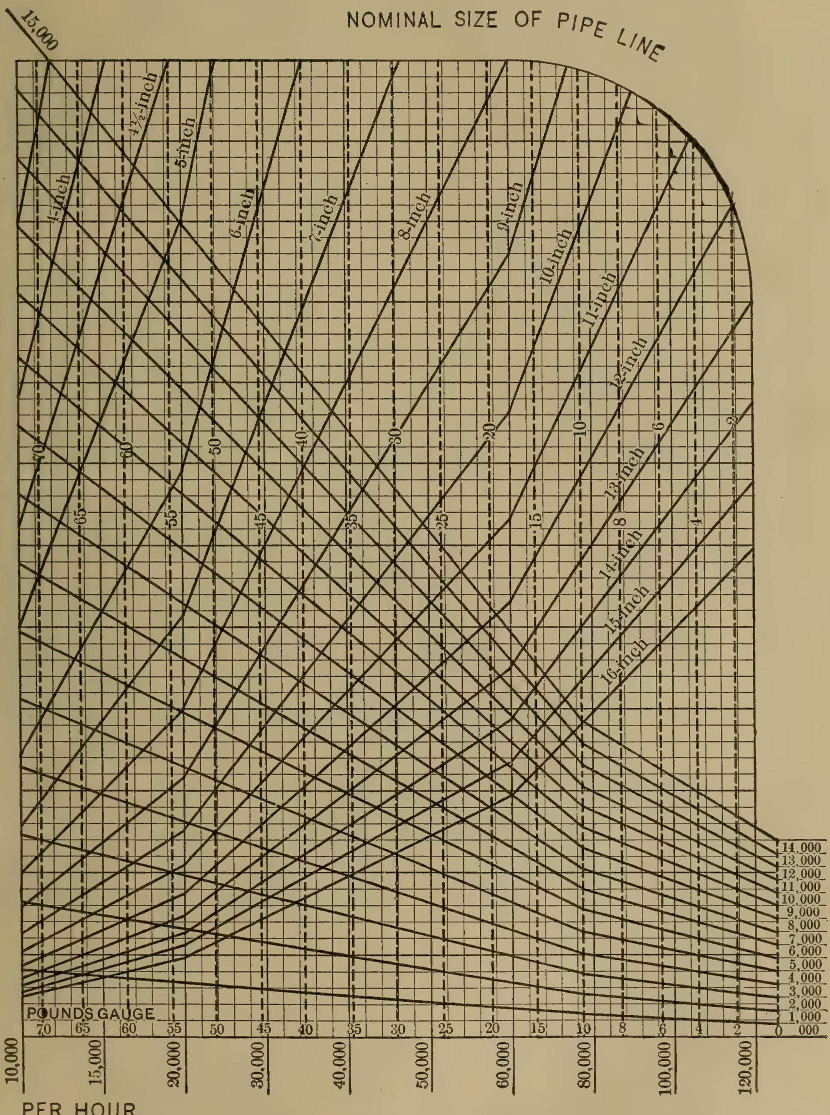

VELOCITY OF STEAM · FEET PER MINUTE
1,000
2,000
3,000
4,000
5,000
6,000
7,000
8,000
9,000
10,000
11,000
12,000
13,000
14,000
1-inch
1¼-inch
1½-inch
2-inch
2½-inch
3-inch
3½-inch
4-inch
4½-inch
5-inch
6-inch
7-inch
8-inch
9-inch
10-inch
STEAM PRESSURE
200 190 180 170 160 150 140 130 120 110 100 95 90 85 80 75
500 1,000 1,500 2,000 3,000 4,000 5,000 6,000 8,000 10,000
POUNDS OF STEAM

Fig. 485.



15,000
NOMINAL SIZE OF PIPE LINE
4-inch
4½-inch
5-inch
6-inch
7-inch
8-inch
9-inch
10-inch
11-inch
12-inch
13-inch
14-inch
15-inch
16-inch
70
65
60
55
50
45
40
35
30
25
20
15
10
8
6
4
2
14,000
13,000
12,000
11,000
10,000
9,000
8,000
7,000
6,000
5,000
4,000
3,000
2,000
1,000
000
0
POUNDS GAUGE
10,000
15,000
20,000
30,000
40,000
50,000
60,000
80,000
100,000
120,000
PER HOUR

FIG. 485.

Table 116 gives a few of the best-known formulas for the flow of steam, and Table 117 a comparison between them with respect to velocity, weight discharged, diameter, and the drop in pressure.

Formula 11, Table 116, is the most commonly accepted, and the curves in Fig. 484 are based upon it, assuming a steam pressure of 100 pounds absolute and pipe lengths of 100 feet. Within the limit of 12,000 feet per minute velocity and 10 pounds per square inch drop in pressure the curves are sufficiently accurate for all practical purposes, but beyond this range the results are purely conjectural and may not be accurate, as no recorded experiments have been conducted at these high velocities or with pipes of large diameters.

Though applicable directly to pipes 100 feet long with mean pressure of 100 pounds per square inch absolute, they may be used for any length or pressure. For example, for any length other than 100 feet, multiply the drop given in the curves by the required length in feet and divide by 100. For any pressure other than 100 pounds absolute, divide the drop given in the curves by 0.2271 (density of steam in pounds) and multiply by the density of steam at the required pressure.

Table 118 is the table ordinarily used in connection with the flow of steam and is calculated from equation 11. Table 119 is based upon equations (4) to (12). The results differ slightly from those in Table 118, though the latter is more comprehensive. The left-hand half of Table 119 gives the discharge in pounds per minute for pipes of various diameters corresponding to drop of pressure as given on the right-hand side in the same horizontal line; e.g., a 6-inch pipe 100 feet long discharges 371 pounds of steam per minute for a drop of 16.4 pounds at 100 pounds pressure.

The curves in Fig. 485 offer a simple means of calculating velocities, discharge, and size of pipe for various conditions of flow. The curves are plotted for saturated steam only. For superheated or moist steam substitute from steam tables pressures corresponding to densities and follow chart as directed.

Example: Allowing a velocity of 5000 feet per minute, what size of pipe is necessary to deliver 8000 pounds of steam per hour at 120 pounds gauge pressure?

Trace 5000 feet velocity line to 120 pounds gauge line. From intersection follow horizontally to "8000 pounds of steam per hour" and read nearest size of pipe, viz., 4 inches.

Example: Find velocity of steam in a 6-inch pipe delivering 20,000 pounds of steam per hour at 85 pounds gauge pressure.

Trace the line representing 20,000 pounds per hour until it inter-

TABLE 118.

FLOW OF STEAM THROUGH PIPES (BABCOCK).

Initial Pressure by Gauge. Pounds per Square Inch.	Diameter of Pipe, in Inches. Length of each = 240 diameters.						
	$\frac{3}{4}$	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	4
	Weight of Steam per Minute, in pounds, with One Pound Loss of Pressure.						
1	1.16	2.07	5.7	10.27	15.45	25.38	46.85
10	1.44	2.57	7.1	12.72	19.15	31.45	58.05
20	1.70	3.02	8.3	14.94	22.49	36.94	68.20
30	1.91	3.40	9.4	16.84	25.35	41.63	76.84
40	2.10	3.74	10.3	18.51	27.87	45.77	84.49
50	2.27	4.04	11.2	20.01	30.13	49.48	91.34
60	2.43	4.32	11.9	21.38	32.19	52.87	97.60
70	2.57	4.58	12.6	22.65	34.10	56.00	103.37
80	2.71	4.82	13.3	23.82	35.87	58.91	108.74
90	2.83	5.04	13.9	24.92	37.52	61.62	113.74
100	2.95	5.25	14.5	25.96	39.07	64.18	118.47
120	3.16	5.63	15.5	27.85	41.93	68.87	127.12
150	3.45	6.14	17.0	30.37	45.72	75.09	138.61

Initial Pressure by Gauge. Pounds per Square Inch.	Diameter of Pipe, in Inches. Length of each = 240 diameters.						
	5	6	8	10	12	15	18
	Weight of Steam per Minute, in Pounds, with One Pound Loss of Pressure.						
1	77.3	115.9	211.4	341.1	502.4	804	1177
10	95.8	143.6	262.0	422.7	622.5	996	1458
20	112.6	168.7	307.8	496.5	731.3	1170	1713
30	126.9	190.1	346.8	559.5	824.1	1318	1930
40	139.5	209.0	381.3	615.3	906.0	1450	2122
50	150.8	226.0	412.2	665.0	979.5	1567	2294
60	161.1	241.5	440.5	710.6	1046.7	1675	2451
70	170.7	255.8	466.5	752.7	1108.5	1774	2596
80	179.5	269.0	490.7	791.7	1166.1	1866	2731
90	187.8	281.4	513.3	828.1	1219.8	1951	2856
100	195.6	293.1	534.6	862.6	1270.1	2032	2975
120	209.9	314.5	573.7	925.6	1363.3	2181	3193
150	228.8	343.0	625.5	1009.2	1486.5	2378	3481

For any other length divide 240 by the given length expressed in diameters and multiply the tabular quantity by the square root of this quotient, which will give the flow for one pound loss of pressure. Conversely, dividing the given length by 240 will give the loss of pressure for the flow given in the table.

TABLE 119.

FLOW OF STEAM THROUGH PIPES (SICKLES).

Length of Pipe One Thousand Feet.

Discharge in Pounds per Minute Corresponding to Drop in Pressure on Right for Pipe Diameters in Inches in Top Line.											Drop in Pressure in Pounds per Square Inch Corresponding to Discharge on Left; Densities and Corresponding Absolute Pressures per Square Inch in First Two Lines.								
Diameter.	12″	10″	8″	6″	4″	3″	2½″	2″	1½″	1″	Density. Pressure.	0.208 90	0.230 100	0.284 125	0.328 150	0.401 180	0.443 200	0.506 230	0.548 250
Discharge	2328	1443	799	371	123.	55.9	28.8	18.1	6.81	2.52	Drop.	18.10	16.4	13.3	11.1	9.39	8.50	7.44	6.87
Do....	2165	1341	742	344	114.6	51.9	27.6	16.8	6.52	2.34	Do..	15.60	14.1	11.4	9.60	8.09	7.33	6.41	5.92
Do....	1996	1237	685	318	106.	47.9	26.4	15.5	6.24	2.16	Do..	13.3	12.0	9.74	8.18	6.90	6.24	5.47	5.05
Do....	1830	1134	628	292	97.0	43.9	25.2	14.2	5.95	1.98	Do..	11.1	10.0	8.13	6.83	5.76	5.21	4.56	4.21
Do....	1663	1031	571	265	88.2	39.9	24.0	12.9	5.67	1.80	Do..	9.25	8.36	6.78	5.69	4.80	4.34	3.80	3.51
Do....	1580	979	542	252	83.8	37.9	22.8	12.3	5.29	1.71	Do..	8.33	7.53	6.10	5.13	4.32	3.91	3.42	3.16
Do....	1497	928	514	239	79.4	35.9	21.6	11.6	5.00	1.62	Do..	7.48	6.76	5.48	4.60	3.88	3.51	3.07	2.84
Do....	1414	876	485	226	75.	33.9	20.4	10.9	4.72	1.53	Do..	6.67	6.03	4.88	4.10	3.46	3.13	2.74	2.53
Do....	1331	825	457	212	70.6	31.9	19.2	10.3	4.43	1.44	Do..	5.91	5.35	4.33	3.64	3.07	2.78	2.43	2.24
Do....	1248	873	428	199	66.2	28.9	18.0	9.68	4.15	1.35	Do..	5.19	4.69	3.80	3.19	2.69	2.44	2.13	1.97
Do....	1164	722	400	186	61.7	27.9	16.8	9.03	3.86	1.26	Do..	4.52	4.09	3.31	2.78	2.34	2.12	1.86	1.72
Do....	1081	670	371	172	57.3	25.9	15.6	8.38	3.68	1.17	Do..	3.90	3.53	2.86	2.40	2.02	1.83	1.60	1.48
Do....	998	619	343	159	52.9	23.9	14.4	7.74	3.40	1.08	Do..	3.32	3.00	2.43	2.04	1.72	1.56	1.36	1.26
Do....	915	567	314	146	48.5	21.9	13.2	7.10	3.11	0.99	Do..	2.79	2.52	2.04	1.72	1.45	1.31	1.15	1.06
Do....	832	516	286	132	44.1	20.0	12.0	6.45	2.83	0.90	Do..	2.31	2.09	1.69	1.42	1.20	1.08	0.949	0.877
Do....	748	464	257	119	39.7	18.0	10.8	5.81	2.55	0.81	Do..	1.87	1.69	1.37	1.15	0.97	0.878	0.769	0.710
Do....	665	412	228	106	35.3	16.0	9.6	5.16	2.26	0.72	Do..	1.47	1.33	1.08	0.905	0.762	0.690	0.604	0.558
Do....	582	361	200	92.8	30.9	14.0	8.4	4.52	1.98	0.63	Do..	1.13	1.02	0.828	0.695	0.586	0.531	0.456	0.429

To get the pressure drop for lengths other than 1000 feet, multiply by lengths in feet ÷ 1000.

sects "6-inch pipe" line. Follow horizontally to "85 pounds gauge" and read 7350 feet per minute.

Example: Allowing a velocity of 6000 feet per minute through an 8-inch pipe, find the pounds of steam flowing per hour at 100 pounds gauge.

Trace the "6000 feet per minute" velocity line until it intersects "100 pounds pressure" line. Follow horizontally to 8-inch pipe line and read 32,200 pounds.

377. Equation of Pipes. — It is frequently desirable to know what number of one sized pipes will be equal in capacity to another pipe.

According to the formulas in Group II, Table 117, the weights discharged vary with the square root of the fifth power of the diameter; that is, the number of pipes equal in capacity to any given pipe may be determined from the equation

$$N_1 = d^{\frac{5}{2}} \div d_1^{\frac{5}{2}}, \tag{257}$$

in which N_1 = number of pipes of diameter d_1 equal in capacity to a pipe of diameter d; d_1 and d in inches.

According to the formulas in Group I, Table 117, the weights discharged vary as $\left\{ d^5 \div \left(1 + \frac{3.6}{d}\right) \right\}^{\frac{1}{2}}$ and the equation becomes

$$N_1 = \left(\frac{d^5}{1 + \frac{3.6}{d}} \div \frac{d_1^5}{1 + \frac{3.6}{d_1}} \right)^{\frac{1}{2}} \tag{258}$$

$$= \left(\frac{d^6 (d_1 + 3.6)}{d_1^6 \, (d + 3.6)} \right) \tag{259}$$

$$= \frac{d^3 \sqrt{d_1 + 3.6}}{d_1^3 \sqrt{d + 3.6}} \cdot \tag{260}$$

From (257) and (260) we see that the values of N_1 are practically the same for either equation when the ratio of d to d_1 is small and that they differ widely for large ratios. For example, according to (257), 5.7 eight-inch pipes are equivalent in capacity to one sixteen-inch pipe, whereas (259) gives 6.15. The difference is negligible. Again, according to (257), 180 two-inch pipes are equivalent in capacity to one sixteen-inch pipe, whereas (260) gives 274. The difference is considerable. Equation (260) is most commonly accepted and is the basis of Table 120.

378. Friction through Valves and Fittings. — The formulas outlined in Table 116 are strictly applicable only to well-lagged pipes, free from bends or obstructions of any kind such as valves or fittings, which greatly increase the resistance of the flow of steam. If these

TABLE 120.

TABLE OF EQUATION OF PIPES. — STANDARD STEAM AND GAS PIPES.

Dia.	½	¾	1	1½	2	2½	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	Dia.
½	......	2.27	4.88	15.8	31.7	52.9	96.9	205	377	620	918	1,292	1,767	2,488	3,014	3,786	4,904	5,927	7,321	8,535	9,717	½
¾	2.60	......	2.05	6.97	14.0	23.3	42.5	90.4	166	273	405	569	779	1,096	1,328	1,668	2,161	2,615	3,226	3,761	4,282	¾
1	7.55	2.90	......	3.45	6.82	11.4	20.9	44.1	81.1	133	198	278	380	536	649	815	1,070	1,263	1,576	1,837	2,092	1
1½	24.2	9.30	3.20	......	2.00	3.34	6.13	13.0	23.8	39.2	58.1	81.7	112	157	190	239	310	375	463	539	614	1½
2	54.8	21.0	7.25	2.26	......	1.67	3.06	6.47	11.9	19.6	29.0	40.8	55.8	78.5	95.1	119	155	187	231	269	307	2
2½	102	39.4	13.6	4.23	1.87	......	1.83	3.87	7.12	11.7	17.4	24.4	33.4	47.0	56.9	71.5	92.6	112	138	161	184	2½
3	170	65.4	22.6	7.03	3.11	1.66	......	2.12	3.89	6.39	9.48	13.3	20.9	23.7	31.2	39.1	50.6	61.1	75.5	88.0	100	3
4	376	144	49.8	15.5	6.87	3.67	2.21	......	1.84	3.02	4.48	6.30	8.61	12.1	14.7	18.5	23.9	28.9	35.7	41.6	47.4	4
5	686	263	90.9	28.3	12.5	6.70	4.03	1.83	......	1.65	2.44	3.43	4.69	6.60	8.00	10.0	13.0	15.7	19.4	22.6	25.8	5
6	1,116	429	148	46.0	20.4	10.9	6.56	2.97	1.63	......	1.48	2.09	2.85	4.02	4.86	6.11	7.91	9.56	11.8	13.8	15.6	6
7	1,707	656	226	70.5	31.2	16.6	10.0	4.54	2.49	1.51	......	1.41	1.93	2.71	3.28	4.12	5.34	6.45	7.97	9.31	10.6	7
8	2,435	936	322	101	44.5	23.8	14.3	6.48	3.54	2.18	1.43	......	1.35	1.93	2.33	2.92	3.79	4.57	5.67	6.60	7.52	8
9	3,335	1,281	440	137	60.8	32.5	19.5	8.85	4.85	2.98	1.95	1.37	......	1.41	1.71	2.14	2.77	3.35	4.14	4.83	5.50	9
10	4,393	1,688	582	181	80.4	42.9	25.8	11.7	6.40	3.93	2.57	1.80	1.32	.....	1.21	1.52	1.97	2.38	2.94	3.43	3.91	10
11	5,642	2,168	747	233	103	55.1	33.1	15.0	8.22	5.05	3.31	2.32	1.70	1.28	.....	1.26	1.63	1.88	2.43	2.83	3.22	11
12	7,087	2,723	938	293	129	69.2	41.6	18.8	10.3	6.34	4.15	2.91	2.13	1.61	1.26	.....	1.30	1.57	1.93	2.26	2.58	12
13	8,657	3,326	1,146	358	158	84.5	50.7	23.0	12.6	7.75	5.07	3.56	2.60	1.98	1.53	1.22	.....	1.21	1.49	1.74	1.98	13
14	10,600	4,070	1,403	438	193	103	62.2	28.2	15.4	9.48	6.21	4.35	3.18	2.41	1.88	1.50	1.22	.....	1.24	1.44	1.64	14
15	12,824	4,927	1,698	530	234	125	75.3	34.1	18.7	11.5	7.52	5.27	3.85	2.92	2.27	1.81	1.48	1.21	.....	1.17	1.35	15
16	14,978	5,758	1,984	619	274	146	88.0	39.9	21.8	13.4	8.78	6.15	4.51	3.41	2.66	2.12	1.73	1.42	1.18	.....	1.14	16
17	17,537	6,738	2,322	724	320	171	103	46.6	25.6	15.7	10.3	7.20	5.27	3.99	3.11	2.47	2.03	1.66	1.37	1.17	.....	....
18	20,327	7,810	2,691	840	371	198	119	54.1	29.6	18.2	11.9	8.35	6.11	4.63	3.60	2.87	2.35	1.92	1.59	1.36	1.16	....
20	26,676	10,249	3,532	1,102	487	260	157	70.9	38.9	23.9	15.6	10.9	8.02	6.07	4.73	3.76	3.08	2.52	2.08	1.78	1.52	....
24	42,624	16,376	5,644	1,761	778	416	250	113	62.1	38.2	25.0	17.5	12.8	9.70	7.55	6.01	4.92	4.02	3.32	2.84	2.43	....
30	75,453	28,990	9,990	3,117	1,378	736	443	201	110	67.6	44.2	31.0	22.7	17.2	13.4	10.7	8.72	7.14	5.88	5.03	4.30	....
36	120,100	46,143	15,902	4,961	2,193	1,172	705	319	175	108	70.4	49.3	36.1	27.3	21.3	16.9	13.9	11.3	9.37	8.01	6.85	....
42	177,724	68,282	23,531	7,341	3,245	1,734	1,044	473	259	159	104	73.0	53.4	40.5	31.5	25.1	20.5	16.8	13.9	11.9	10.1	....
48	249,351	95,818	33,020	10,301	4,554	2,434	1,465	663	363	223	146	102	75.0	56.8	44.2	35.2	28.8	23.5	19.4	16.6	14.2	....
Dia.	½	¾	1	1½	2	2½	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	

The upper portion above the diagonal line of blanks pertains to "standard" sizes while the lower portion is for pipe of the actual internal diameter given.

obstructions must be considered, it is customary to allow for them by assuming an added length of straight pipe equivalent in resistance to the various fittings and bends. Unfortunately, the few tests which have been made for the purpose of determining the resistance of various pipe fittings give discordant results, and in the absence of more recent data the rules given by Briggs ("Warming Buildings by Steam") are probably as accurate as any.

According to Briggs, the length of pipe in inches equivalent to the resistance of one standard 90-degree elbow is

$$L = 76\,d \div \left(1 + \frac{3.6}{d}\right) \tag{261}$$

and to that of one globe valve

$$L = 114\,d \div \left(1 + \frac{3.6}{d}\right). \tag{262}$$

The resistance of gate valves is not considered.

379. Exhaust Piping, Condensing Plants. — The exhaust piping in condensing plants is arranged either according to (1) the *independent* or (2) the *central condensing system.* In the former each engine is provided with an independent condenser and air pump. In case the vacuum "drops" or it is desired to operate non-condensing, the steam is discharged through a branch pipe with relief valve to the atmosphere, Figs. 3 and 326. When there are a number of engines in one installation the atmospheric pipes lead to a common *free exhaust main,* which, on account of its great size, is ordinarily constructed of light-weight riveted steel pipe. The short connection between engine and condenser is usually made with lap-welded steel pipe, since riveted joints are apt to leak, due to the engine vibrations. In a central condensing plant, Fig. 333, the several engines exhaust through a common main into a single large condenser. An atmospheric relief valve is usually provided in connection with the condenser, and no free exhaust main is necessary. Several arrangements of condenser piping are illustrated in Figs. 326 to 333.

380. Exhaust Piping, Non-condensing Plant. Webster Vacuum System. — In the majority of non-condensing plants the exhaust steam is used for heating purposes. One of the best-known systems of exhaust steam heating, in which the back pressure on the engine is reduced by circulating below atmospheric pressure, is that known as the *Webster combination system.* The general arrangement is illustrated in Fig. 2 and the principles of operation are described in paragraph 3. It has the advantage of affording (1) minimum back pressure on the engine; (2) effective and continuous drainage of condensation from supply pipes

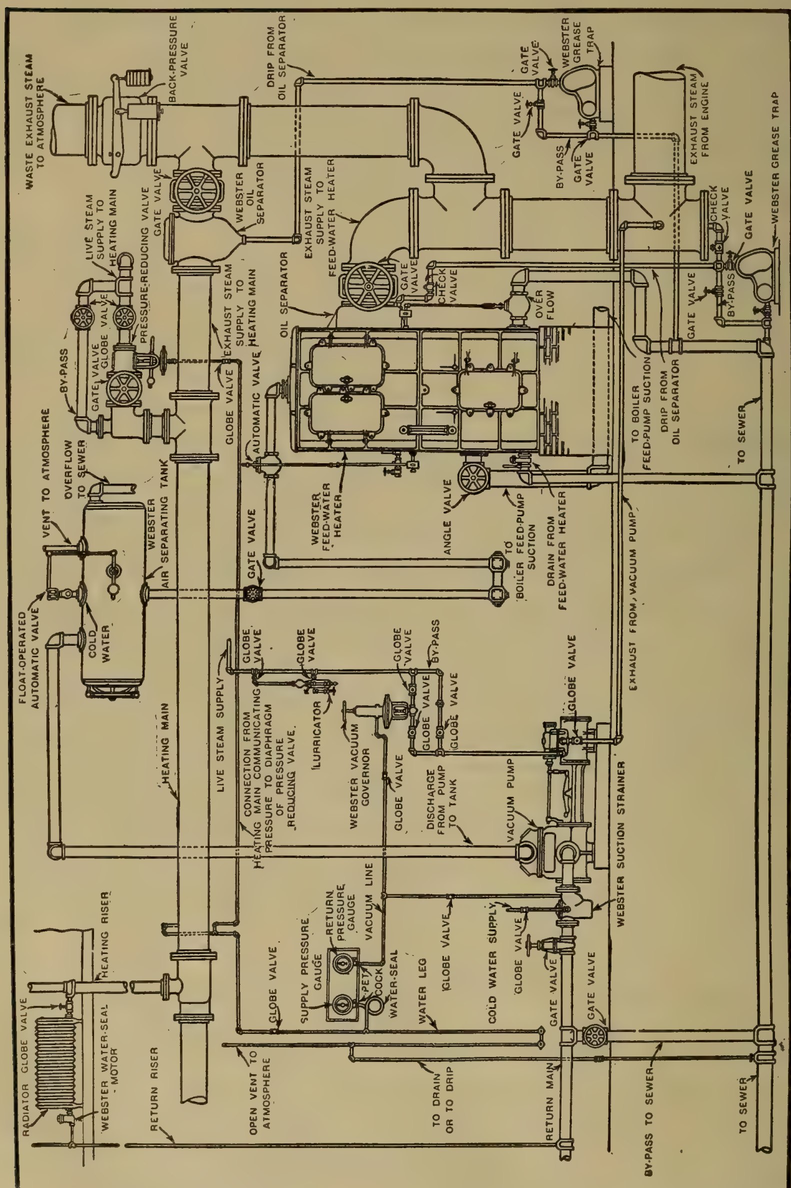


FIG. 486. Diagrammatic Arrangement of Webster Vacuum Heating System.

and radiators; (3) continuous removal of air and entrained moisture from confined spaces; (4) independent regulation of temperature in each radiator; (5) continuous return of condensation to the boiler; (6) utilization of part of the exhaust steam for preheating the feed water; and (7) automatic regulation. Fig. 486 gives a diagrammatic arrangement of the piping and appurtenances in a typical installation. The characteristic feature of this system is the automatic outlet valve attached to each part requiring drainage, which permits both the water of condensation and the non-condensable gases to be removed continuously. The radiator temperature may be regulated by varying the quantity of steam supplied, either by hand or automatically by thermostatic control. The Webster valve, Fig. 487, enables the vacuum to withdraw the water of condensation as fast as it is formed irrespective

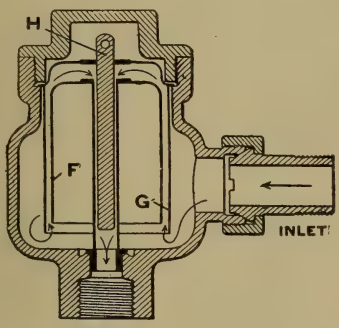


FIG. 487. Webster Air Valve.

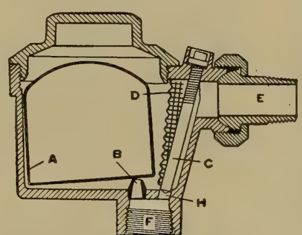


FIG. 488. Automatic Vacuum Valve, Illinois Engineering Co.

of the pressure in the radiator; hence the supply may be throttled to such an extent that the temperature in the radiator is practically as low as that of steam corresponding to the pressure in the vacuum line. The small annular space between the inner tube of the float F and the guide H permits of a vacuum in the body of the valve. When the water from the radiator lifts the float the water is drawn into the returns pipe. The valve then returns to its seat and the escape of steam is prevented, except such as finds its way through the annular space around the guide stem H. An improvement on this valve which prevents the escape of steam is illustrated in Fig. 488. When steam is admitted to the radiator the condensation flows into the valve, righting the float A and sealing the outlet B against the passage of steam; as the valve fills with water the buoyancy of the float raises it from its seat and permits the water to be drawn out; the float falls and reseats on the nipple when about a half-inch of water remains in the valve, thus maintaining a water seal.

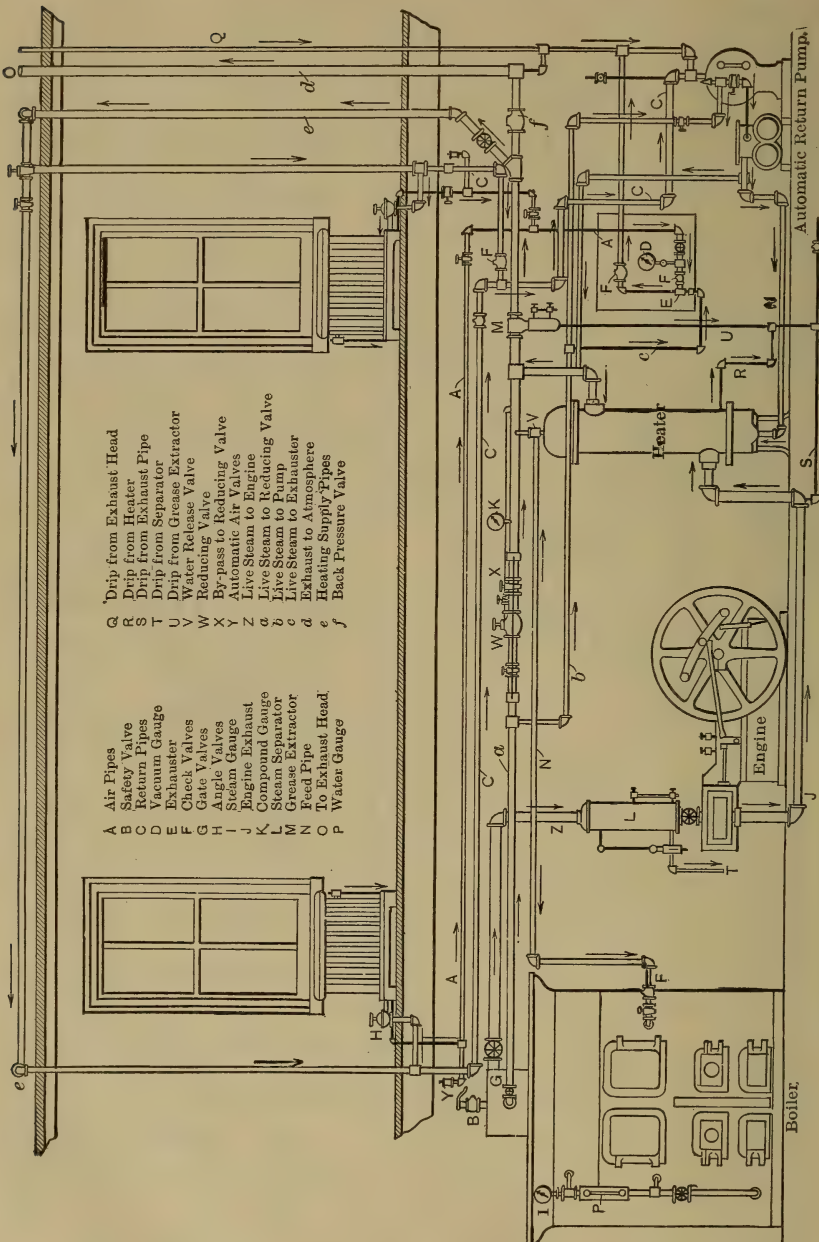


FIG. 489. Diagrammatic Arrangement of Piping in the Paul Vacuum Heating System.

Screen *D* prevents scale and dirt from entering the valve proper. By-pass *H* is for emergency use in draining off accumulated water in the radiator in case the valve becomes stopped up, and permits the bonnet to be removed without trouble from the accumulated water.

381. Exhaust Piping, Non-condensing Plants. Paul Heating System. — The Paul vacuum system differs from the Webster in that the condensation, and the air and non-condensable gases are separately handled. Referring to Fig. 489, which gives a diagrammatic arrangement of the piping, the condensed steam gravitates to the *automatic returns tank and pump* and is pumped either directly to the boiler or through the heater to the boiler. Air and vapor are withdrawn from the upper part of the radiator by the *Paul exhauster* or ejector *E*, and discharged into the returns tank, which is vented to the atmosphere for the escape of the non-condensable gases. The exhauster receives its supply of steam through pipe *O*, Fig. 490, which shows the general arrangement of this apparatus. The piping is in duplicate to guard against failure to operate. The suction side of the exhauster is connected with the air pipes *A*, *A*, Fig. 489. Fig. 491 gives a section through the *Paul air* or *vacuum* valve which prevents steam from blowing into the air pipes and permits only air to pass. In Fig. 489 the heating system is piped on what is known as the "one-pipe down-feed" principle; i.e., the exhaust steam is first conducted to a distributing header in the attic, from which the various supply pipes are led to the radiators. The water of condensation

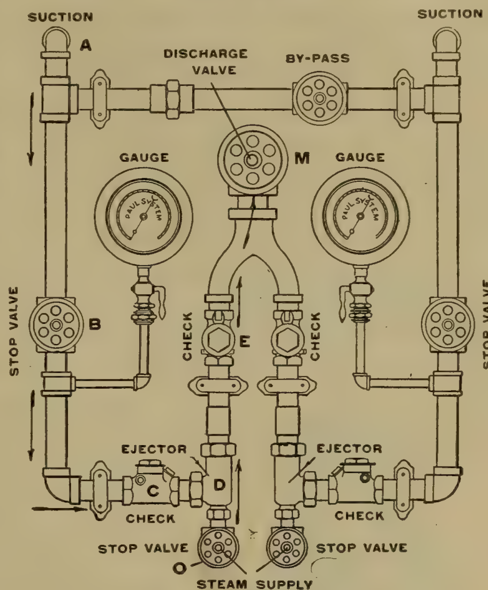


FIG. 490. Paul Exhauster.

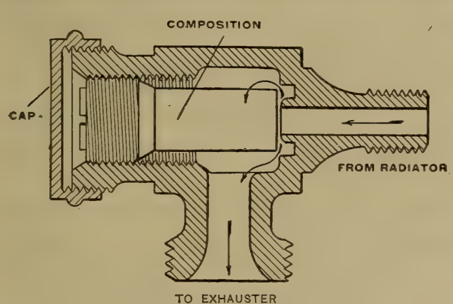


FIG. 491. Paul Vacuum Valve.

returns through these same pipes and gravitates to the returns pump. Both the supply steam and the condensation flow in the same direction. This system is also piped on the "one-pipe up-feed," the "two-pipe up-feed," and the "two-pipe down-feed" principle. The "one-pipe up-feed" differs from the system just described in that the steam flows upward through the risers and does away with the attic piping. The returns, however, flow against the current of steam, and water hammer is more likely to occur than with the down-feed system. In the two-pipe systems the steam supply pipes or risers conduct steam only, and the returns carry the condensation. The one-pipe down-feed is cheaper and simpler and practically as efficient as the two-pipe system under normal conditions. It is objectionable, however, due to the difficulty of draining the radiator with closely throttled supply valve, since the velocity of the entering steam prevents the water from returning through the same orifice.

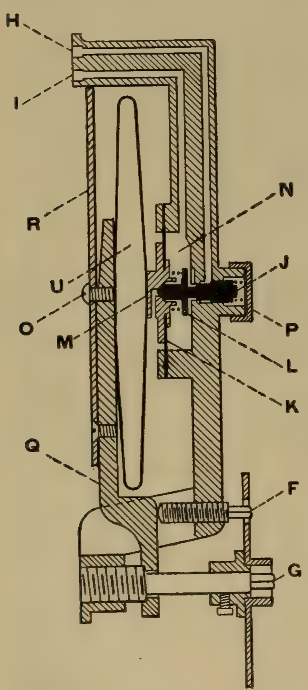


FIG. 492. Section through Powers Thermostat.

382. Automatic Temperature Control. — Experience shows that a considerable saving in fuel may be effected in the heating plants of tall office buildings and similar plants by automatically controlling the temperature. Hand-controlled valves are usually left wide open, and when the room becomes too hot the temperature is frequently lowered by opening the window, resulting in a waste of heat which may be considerable in modern buildings with hundreds of offices. Many successful methods of automatic temperature control are available, the usual system consisting of *thermostats* which control the supply of heat by means of *diaphragm valves*, the latter taking the place of the usual radiator supply valve.

Fig. 492 shows a Powers thermostat. The expansible disk U contains a volatile liquid having a boiling point of about 50 degrees F. The pressure of the vapor within the disk at a temperature of 70 degrees amounts to six pounds to the square inch, and varies with every change of temperature, causing a variation in the thickness of the disk. The disk is attached by a single screw O to the lever Q, which rests upon the screw F as a fulcrum. The flat spring R holds the lever and disk against the movable flange M. Connecting with the chamber N are

two air passages H and I. The thermostat is attached by means of two screws at the upper end to a wall plate permanently secured to the wall. This wall plate has ports registering with H and I, one for supplying air under pressure and the other for conducting it to the diaphragm motor which operates the valve or damper. Air is admitted through H under a pressure of about fifteen pounds per square inch, and its passage into chamber N is regulated by the valve J, which is normally held to its seat by a coil spring under cap P. K is an elastic diaphragm carrying the flange M, with escape valve passage covered by the point of valve L. Valve L tends to remain open by reason of the spring. When the temperature rises sufficiently expansion of the disk U first causes the valve to seat, its spring being weaker than that above valve J. If the expansive motion is continued, valve J is lifted from its seat and compressed air flows into chamber N, exerting a pressure upon the elastic diaphragm K in opposition to the expansive force of the disk. If the temperature falls, the disk contracts and the overbalancing air pressure in N results in a reverse movement of the flange M, permitting the escape valve to open and discharge a portion of the air; thus the air pressure is maintained always in direct proportion to the expansive power (and temperature) of the disk U. The passage I communicates with a diaphragm valve, Fig. 493. The compressed air operates the diaphragm against a coiled spring resistance, so that the movement is proportional to the air pressure and the supply of steam controlled accordingly. The adjusting screw G, squared to receive a key, carries an indicator by means of which the thermostat can be set to carry any desired temperature within its range, usually from 60 to 80 degrees. In changing the temperature adjustment lever Q forces the disk U closer to or farther from the flange M.

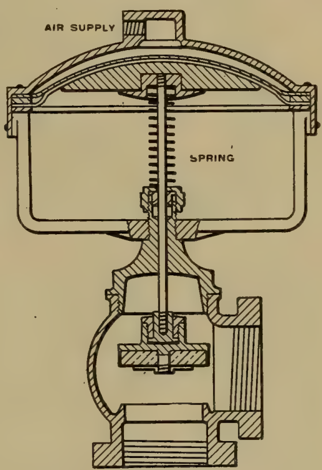


FIG. 493. A Typical Diaphragm Valve.

In connecting up the system compressed air is carried to the thermostat and diaphragm valves, from a reservoir through small concealed pipes.

In the indirect system of heating the dampers are of the diaphragm type and the method of regulation is the same as with the direct system.

383. Feed-water Piping. — The simplest arrangement of feed-water piping may be found in non-condensing plants, in which the feed water

is obtained under a slight head, such as is afforded by the average city supply, and is heated in an open heater by the exhaust steam from the engine to a temperature varying from 180 to 210 degrees F. The hot feed water gravitates from the heater to the pump and then is forced to the boiler, or to the economizer if one is used. If a meter is used

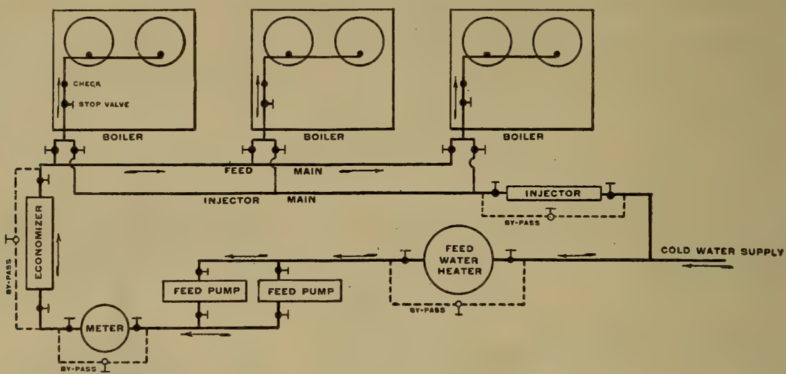


FIG. 494. Feed-water Piping; Non-condensing Plant.

it is generally placed on the discharge side of the pump, and should be by-passed to permit it to be cut out for repairs (Fig. 494). Plants operating continuously should have feed pumps in duplicate. In some cases the returns from the heating system gravitate to the heater and only enough cold water is added to make up the loss from leakage, etc.

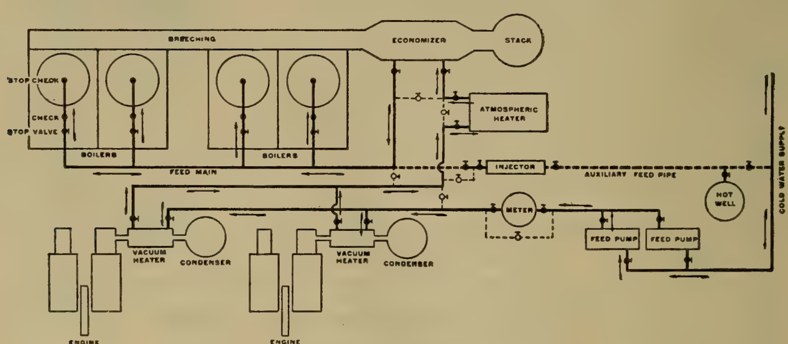


FIG. 495. Feed-water Piping; Condensing Plant.

In other cases the returns gravitate to a special "returns tank," from which they are pumped directly to the boiler without further heating. Occasionally a live-steam purifier is used, especially if the water contains a large percentage of calcium sulphate. The feed is then subjected to boiler pressure and temperature and the greater part of the impurity precipitated before it enters the boiler. Closed heaters are often used

in place of open heaters. When the supply is not under head a closed heater is usually preferred and is placed between the pump discharge and the feed main.

In condensing plants the feed piping is similar to that in non-condensing plants, except that if exhaust steam is used for heating purposes it is supplied by the auxiliaries, such as feed pumps, stoker engines, condenser engines, and other steam-using appliances.

In plants having a number of boilers it is customary to run a feed main or header the full length of the boiler room and connect it to each boiler by a branch pipe. This main may be a simple header or in duplicate or of the "loop" or "ring" type. Horizontal tubular boilers are frequently arranged in one battery with the feed main run along the fronts of the boilers just above the fire doors. Water-tube boilers are generally set in a battery, and as the arrangement above would block the passageway between the batteries, the main is run either above or under the settings, the former being the more common. Where a single header is used, the feed pumps are sometimes placed so as to feed into opposite ends of the main, which is then cut into sections by valves. Another arrangement is to place the pumps so as to feed into the middle of the header. With the loop arrangement the main is ordinarily cut into sections by valves so that the water may be sent either way from the pumps and any defective section cut out. With duplicate mains a common arrangement is to place one main along the front of the boiler and the other at the rear or both overhead as in Fig. 482. Sometimes one main is placed in the passageway below the boiler setting and the other on top.

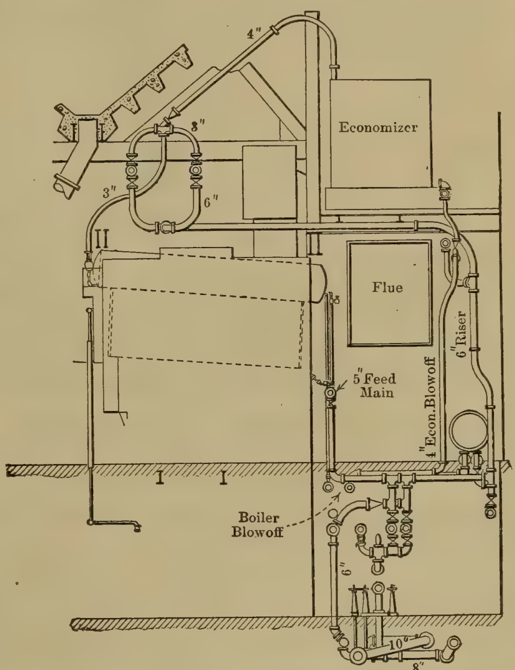


FIG. 496. Feed-water Piping.

Standard wrought-iron pipe is usually used for pressures under 100 pounds and extra heavy pipe for greater pressures. The pipes and

fittings from boiler to main are frequently of brass, and preferably so, since brass withstands corrosive action much better than iron or steel. Flanged joints should be used in all cases, since the pockets formed by the ordinary screwed joints hasten corrosion at those points. (Power, June, 1902, p. 4.)

Fig. 497, *A* to *E*, illustrates the various combinations of check valve, stop valves, and regulating valve in steam boiler practice. The simplest

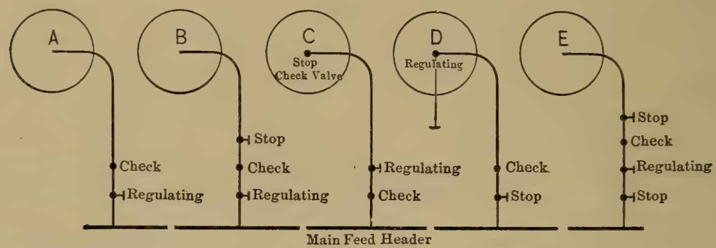


FIG. 497. Different Arrangements of Valves in Feed-water Branch Pipes.

arrangement and one sometimes used in plants operating intermittently is shown in *A*. Here there are but two valves between the boiler and the main, the check being nearest the boiler and the stop valve at the main. The stop valve performs both the function of cutting out the boiler and of regulating the water supply. This arrangement is not recommended, as any sticking or excessive leaking of the check valve will necessitate shutting down the boiler. *B* shows the most common arrangement. Here the check valve is placed between the regulating valve and a stop valve as indicated. This permits a disabled check to be easily removed while pressure is on the boiler and the main. *E* shows an arrangement whereby both check and regulating valve may be removed, and is particularly adapted to boilers operating continuously where the regulating valve is subjected to severe usage. In this case the stop valves are run wide open and are subjected to no wear. The regulating valve most highly recommended is a *self-packing* brass globe valve with *regrinding disk*. The check valve is ordinarily of the *swing check* pattern with regrinding disk, Fig. 508 (C). Modern practice recommends an *automatic water relief valve* in the discharge pipe immediately adjacent to each pump to prevent excessive pressure in case a valve is accidentally closed in by-passing or in changing over.

384. Flow of Water through Orifices, Nozzles, and Pipes. — Bernoulli's theorem is the rational basis of most empirical formulas for the steady flow of a fluid from an up-stream position *n* to a down-stream position *m*, thus ("Mechanics of Engineering," Church, p. 706):

$$\frac{P_m}{\gamma} + \frac{V_m{}^2}{2g} + Z_m = \frac{P_n}{\gamma} + \frac{V^{n2}}{2g} + Z_n - \left\{\begin{array}{l}\text{all losses of head}\\ \text{occurring between}\\ n \text{ and } m\end{array}\right\}, \tag{263}$$

in which

V = velocity in feet per second at the point considered.
P = pressure in pounds per square foot.
Z = potential head in feet of the fluid.
γ = density of the fluid, pounds per cubic foot.
g = acceleration of gravity.

Each loss of head will be of the form $K\frac{V^2}{2g}$ in which K is the *coefficient of resistance* to be determined experimentally. The loss of head due to *skin friction* is expressed:

$$H = 4f\frac{l}{d} \times \frac{v^2}{2g}, \tag{264}$$

in which

f = the coefficient of friction of the fluid in the pipe.
l = length of the pipe in feet.
d = diameter of the pipe in feet.
Other notations as in (263).

Discharge from a circular vertical orifice with sharp corners:

$$Q = CA\sqrt{2gh}, \tag{265}$$

in which

Q = cubic feet per second.
C = coefficient, varying from 0.59 to 0.65 (Merriman, "Treatise on Hydraulics," p. 118).
A = area of the orifice, square feet.
h = head of water in feet.
g = acceleration of gravity = 32.2.

Discharge from short cylindrical nozzles three diameters in length, with rounded entrance ("Mechanics of Engineering," Church, p. 690):

$$Q = 0.815\,A\sqrt{2gh}. \tag{266}$$

Discharge from short nozzles with well-rounded corners and conical convergent tubes, angle of convergence $13\frac{1}{2}$ *degrees* (Church, p. 693):

$$Q = 0.94\,A\sqrt{2gh}. \tag{267}$$

Discharge from cylindrical pipe under 500 *diameters in length* (Church, p. 712):

$$Q = 6.3\sqrt{\frac{d^5h}{(1+0.5)\,d + 4fl}}, \tag{268}$$

in which

f = coefficient of friction.

Other notations as above.

f varies with the nature of the inside surface, the diameter of the pipe, and the velocity of flow.

Discharge through very long cylindrical pipes ("Mechanics of Engineering," Church, p. 715):

$$Q = 3.15\sqrt{\frac{d^5h}{fl}}. \tag{269}$$

TABLE OF THE COEFFICIENT f FOR FRICTION OF WATER IN *CLEAN* IRON PIPES.

(Abridged from Fanning.)

Velocity in Ft. per Sec.	Diam. = ½ in. = .0417 ft.	Diam. = 1 in. = .0834 ft.	Diam. = 2 in. = .1667 ft.	Diam. = 3 in. = .25 ft.	Diam. = 4 in. = .333 ft.	Diam. = 6 in. = .50 ft.	Diam. = 8 in. = .667 ft.
0.1	.0150	.0119	.00870	.00800	.00763	.00730	.00704
0.3	.0137	.0113	850	784	750	720	693
0.6	.0124	.0104	822	767	732	702	677
1.0	.0110	.00950	790	743	712	684	659
1.5	.00959	.00868	.00757	.00720	.00693	.00662	.00640
2.0	.00862	810	731	700	678	648	624
2.5	795	768	710	683	662	634	611
3.0	.00753	.00734	.00692	.00670	.00650	.00623	.00600
4.0	722	702	671	651	631	607	586
6.0	689	670	640	622	605	582	562
8.0	663	646	618	600	587	562	544
12.0	630	614	590	582	560	540	522
16.0	.00618	.00600	.00581	.00570	.00552	.00530	.00513
20.0	615	598	579	566	549	525	508

Velocity in Ft. per Sec.	Diam. = 10 in. = .833 ft.	Diam. = 12 in. = 1.00 ft.	Diam. = 16 in. = 1.333 ft.	Diam. = 20 in. = 1.667 ft.	Diam. = 30 in. = 2.50 ft.	Diam. = 40 in. = 3.333 ft.	Diam. = 60 in. = 5. ft.
0.1	.00684	.00669	.00623	....	....	....	....
0.3	673	657	614	.00578	....	....	....
0.6	659	642	603	567	.00504	.00434	.00357
1.0	643	624	588	555	492	428	353
1.5	.00625	.00607	.00572	.00542	.00482	.00421	.00349
2.0	609	593	559	529	470	416	346
2.5	596	581	548	518	460	410	342
3.0	.00584	.00570	.00538	.00509	.00452	.00407	.00339
4.0	568	553	524	498	441	400	333
6.0	548	534	507	482	430	391	324
8.0	532	520	491	470	422	384	320
12.0	512	500	478	457	412	377	.00313
16.0	.00502	.00491	.00470	.00450	.00406	.00370	....
20.0	498	485	....	....	....	....	....

*Loss of head due to friction in water pipes.** Weisbach's formula is as follows:

$$H = \left(0.0144 + \frac{0.01716}{\sqrt{V}}\right)\frac{LV^2}{5.367\,d}, \tag{270}$$

in which

H = friction head in feet.
V = velocity in feet per second.
L = length of pipe in feet.
d = diameter of pipe in inches.

William Cox (American Machinist, Dec. 28, 1893) gives a simple formula which gives almost identical results:

$$H = \frac{(4\,V^2 + 5\,V - 2)\,L}{1200\,d}. \tag{271}$$

Notations as in (270).

Loss of head due to friction of fittings. Formulas (268) to (271) are based on the flow of water through clean straight cylindrical pipes. Where there are bends, valves, or fittings in the line the flow is decreased on account of the additional resistance.

These frictional losses are conveniently expressed in feet of water, thus:

$$H = C\frac{V^2}{2\,g}, \tag{272}$$

C having the following values:

	Angles.		Class of Valve.		
	45 degrees.	90 degrees.	Gate.	Globe.	Angle.
C	0.182	0.98	0.182	1.91	2.94

Example: Determine the pressure necessary to deliver 200 gallons of water per minute through a 4-inch iron pipe line 400 feet long, fitted with four right-angle elbows and two globe valves. The water is to be discharged into an open tank.

A flow of 200 gallons per minute gives a velocity of

$\frac{200 \times 144}{7.48 \times 60 \times 12.72} = 5$ feet per second (7.48 = number of gallons per cubic foot, and 12.72 = internal area of the pipe, square inches).

From the preceding table, $f = 0.00618$ for $V = 5$.

From (272),

Resistance head of 4 elbows $= 0.98 \times \frac{25}{64.4} \times 4 = 1.52$ feet.

* See also, Friction Formulas for Commercial Pipe, by Ira N. Evans, Power, July 9, 1912, p. 54.

Resistance head of 2 globe valves:

$$1.91 \times \frac{25}{64.4} \times 2 = 1.48 \text{ feet.}$$

Resistance head of all fittings:

$$1.52 + 1.48 = 3 \text{ feet.}$$

Substitute $V = 5$, $L = 400$, and $d = 4$ in (271).

$$H = \left(\frac{4 \times 5^2 + 5 \times 5 - 2}{1200 \times 4}\right) 400$$

$$= 10.25 \text{ feet, resistance head of the pipe.}$$

Total resistance head = 10.25 + 3 = 13.25 feet of water, or 5.75 pounds per square inch.

Example: How many gallons of water will be discharged per minute through above line with initial pressure of 100 pounds per square inch, and what will be the pressure at the discharge end?

Since f depends upon the unknown V, we may put $f = 0.006$ for a first approximation and solve for V; then take a new value of f and substitute again, and so on.

Substitute $f = 0.006$, $d = \frac{4}{12}$, $h = 100 \times 2.3 = 230$, and $l = 400$ in (269):

$$Q = 3.15 \sqrt{\frac{0.33^5 + 230}{0.006 \times 400}}$$

= 1.95 cubic feet per second, corresponding to a velocity of 22 feet per second.

From the preceding table,

$f = 0.00548$ (by interpolation) for $V = 22$ feet per second.

From (272) the friction of 4 elbows and 2 globe valves is found to be 58 feet for $V = 22$.

From (164) a resistance head of 58 feet of water for $V = 22$ is found to be equivalent to 136 feet of straight pipe, thus:

$$58 = \left(\frac{4 \times 22^2 \times 5 \times 22 - 2}{1200 \times 4}\right) L.$$

$$L = 136.$$

Substitute $f = 0.0548$, $l = 400 + 136 = 536$ in (162):

$$Q = 3.15 \sqrt{\frac{0.33^5 \times 230}{0.0058 \times 536}}$$

= 1.74 cubic feet per second, corresponding to a velocity of 19.3 feet per second.

= 780 gallons per minute.

If greater accuracy is necessary determine f and L for $V = 19.3$ and proceed as above.

The total friction head may be determined from (271), thus:

$$H = \left(\frac{4 \times 19.3^2 + 5 \times 19.3 - 2}{1200 \times 4}\right) 536$$
$$= 177 \text{ feet of water}$$
$$= 77 \text{ pounds per square inch.}$$

The pressure at the discharge end will be

$$100 - 77 = 23 \text{ pounds per square inch.}$$

Average power plant practice gives the following maximum velocities of flow in water pipes:

Size of Pipe in Inches.	Velocity, Feet per Minute.	Size of Pipe in Inches.	Velocity, Feet per Minute.
$\frac{1}{8}$ to $\frac{1}{2}$	50	3 to 6	250
$\frac{1}{2}$ to $1\frac{1}{2}$	100	Over 6	300–400
$1\frac{1}{2}$ to 3	200		

385. Stop Valves. — The valves used to control and regulate the flow of fluids are the most important element in any piping system. A good valve should have sufficient weight of metal to prevent distortion under varying temperature and pressure, or under strains due to connection with the piping; the seats should be easily repaired or renewed; there should be no pockets or projections for the accumulation of dirt and scale, and the valve stem should permit of easy and efficient packing. Stop valves are made in such a variety of designs that a brief description will be given of only a few fundamental types.

Fig. 498 shows a section of an ordinary *globe valve,* so called because of the globular form of the casing. This type of valve is the most common in use. Globe valves are designated as (1) *inside screw* and (2) *outside screw,* according as the screw portion of the stem is inside the casting, Fig. 498, or outside, Fig. 499. The top, or bonnet, may be screwed into the body of the valve, Fig. 498, or bolted, Fig. 499. The smaller sizes, three inches and under, are usually of the *screw-top* type and the larger of the *bolt-top* type. Valves with *outside yoke* and *screw* are preferable to others in that they show at a glance whether the valve is open or closed, an advantage in changing from one section to another. The disks are made in a variety of forms, the material depending upon the nature of the fluid to be controlled. Thus, for cold water, hard rubber composition gives good results; for hot water

and low-pressure steam, Babbitt metal; for high-pressure steam, copper or bronze; and for highly superheated steam, nickel. The valve bodies are of brass for sizes under three inches, cast iron for the larger sizes and ordinary pressures and temperatures, and cast steel or semi-steel for high temperatures and pressures. Globe valves should always be set to close against the pressure, otherwise they could not be opened if the valves should become detached from the stem. Globe valves should never be placed in a horizontal steam return pipe with the stem vertical, because the condensation will fill the pipe about half full before

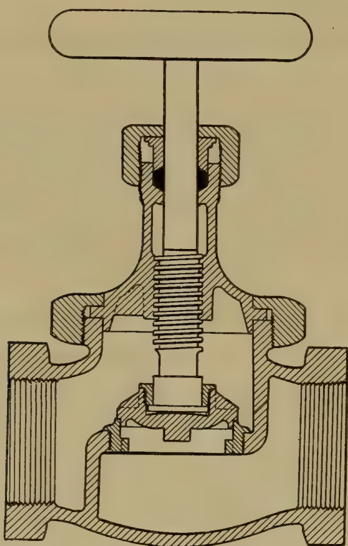


FIG. 498. A Typical Globe Valve, Screw-top, Inside Screw.

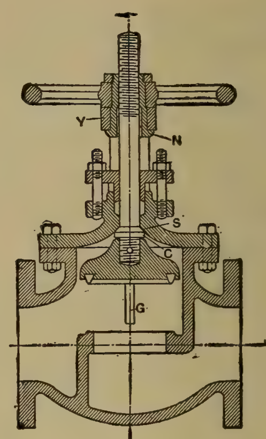


FIG. 499. A Typical Globe Valve, Bolt-top, Outside Screw.

it can flow through the valve. Globe valves that are open all the time are preferably designed with a *self-packing spindle*, as in Fig. 499, in which the top of shoulder C can be drawn tightly against the under surface of bonnet S, thus preventing steam from leaking past the screw threads while the spindle is being packed.

Figs. 500 to 503 show different types of *gate* or *straightway* valves. These valves offer little resistance to the flow of steam or liquid passing through them, and are generally used in the best class of work. Fig. 500 shows a section through a *solid-wedge* gate valve with outside screw and yoke. This form of outside screw and yoke with stem protruding beyond the hand wheel is a perfect indicator to show whether the valve is open or shut, as the hand wheel is stationary and the spindle rises in direct proportion to the amount the valve is opened. For these reasons

outside screw valves are preferable for high-pressure work and especially for the larger sizes. The seats are made solid, or removable, and of various materials for different pressures and temperatures. Fig. 502 shows a section through a *split-wedge* gate valve with parallel faces and seats. For the sake of illustration this valve is fitted with inside screw. In this design the spindle remains stationary so far as any vertical movement is concerned, and the gate or plug, being attached to it by means of a threaded nut, rises into the bonnet when the spindle is revolved. It is impossible to tell by its appearance whether this form of

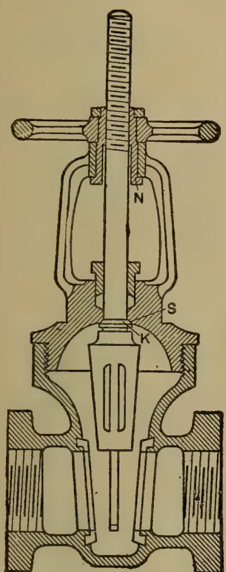


Fig. 500. A Typical Gate Valve, Solid-wedge, Screw-top, Outside Screw.

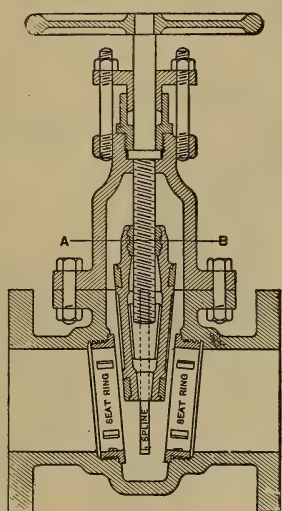


Fig. 501. A Typical Gate Valve, Solid-wedge, Bolt-top, Inside Screw.

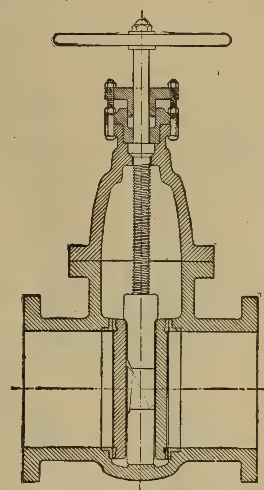


Fig. 502. A Typical Gate Valve, Split-wedge, Bolt-top, Inside Screw.

valve is open or closed. Valves with inside screw are adapted to situations where there is considerable dirt and grit, since the screw is inclosed and protected, and excessive wear is thus avoided. Gate valves with split gates are more flexible than those with solid gates, and hence are less likely to leak. Fig. 503 shows the application of the gate system to an angle valve. All high-pressure valves above 8 inches in diameter should be provided with a small by-pass valve, as the pressure exerted against the disk or gate is very great when the valve is closed and the force required to move it is considerable. The by-pass valve also facilitates "warming up" the section to be cut in and is more readily operated than the main valve.

386. Automatic Non-return Valves. — Fig. 504 shows a section through an *automatic non-return* valve as applied to the nozzle of a steam boiler. As will be seen from the illustration it practically amounts to a large check valve with cushioned disk. The object of this device is the equalization of pressure between the different units of the battery, the valve remaining closed as long as the individual boiler pressure is lower than that of the header. In case a tube blows out the valve closes automatically, owing to the reduction of pressure, and prevents the header steam from entering the boiler. It acts also as a

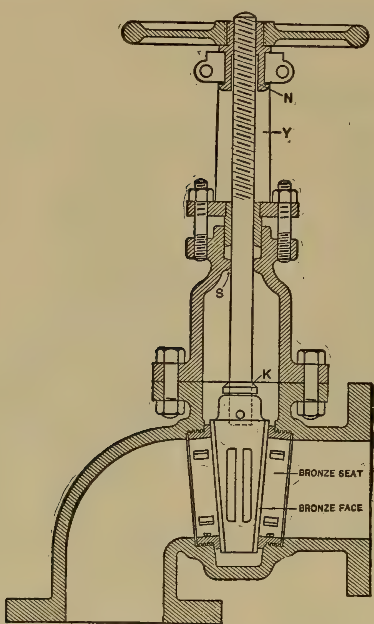


FIG. 503. Ludlow Angle Valve, Gate Pattern.

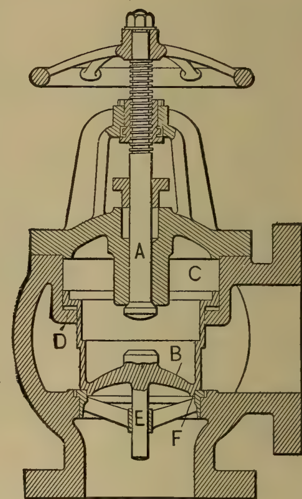


FIG. 504. Anderson Non-return Valve.

safety stop to prevent steam being turned into a cold boiler while men are working inside, because it cannot be opened when there is pressure on the header side only. To be successful, such a valve should not open until the pressure in the boiler is equal to that in the header; it should not stick and become inoperative nor chatter and hammer while performing its work. Referring to Fig. 504, tail rod *E* insures alignment and hence prevents sticking; steam space *C* acts as a dashpot to prevent hammering of the valve as it rises, and steam space *D* acts as a cushion and prevents hammering at closing. Lip *F* is made to enter the opening in the seat and reduce wire drawing across the seat. Fig. 479 shows the installation of a number of non-return valves at the Yonkers power house of the New York Central Railway Company.

387. Emergency Valves and Automatic Stops. — In large power plants it is customary to protect the various divisions of the steam piping by *emergency valves* which may be closed by suitable means at any reasonable distance from the valve. The simplest form of emergency stop is a weighted "butterfly" valve, which is to all intents and purposes a weighted check, as illustrated in Fig. 508 (D). The weight when supported, say by a cord and pulley, holds the valve open; when the cord is cut or released the weight drops and forces the valve shut. The cord may lead to any convenient and safe distance from the valve. In applying this system of control to steam engines the valve is placed in the steam pipe just above the throttle and the weight

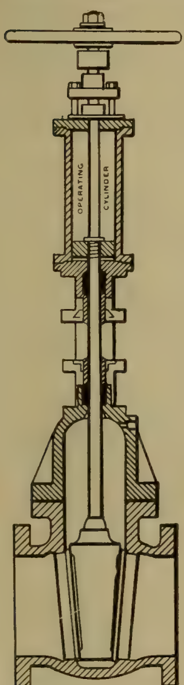


Fig. 505. Crane Emergency Valve, Hydraulic.

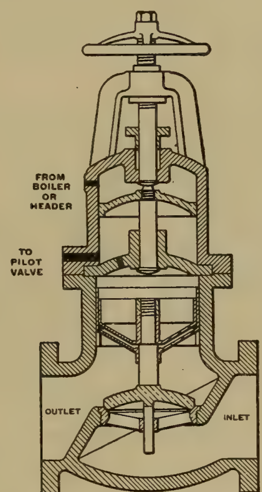


Fig. 506. Anderson Triple-duty Emergency Valve.

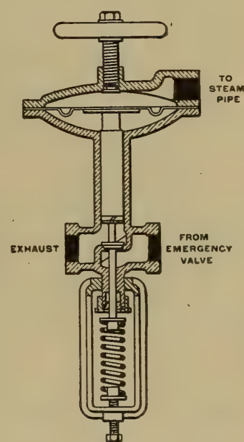


Fig. 507. Pilot Valve for Anderson Triple-duty Emergency Valve.

held up by a lever controlled by the main governor or preferably by a separate governor. Should the engine exceed a certain speed, as in case of accident to the regular governor, the lever supporting the weight is tripped by the emergency governor and the valve is closed automatically. For high pressures a rotating plug valve or cock is preferred to the butterfly type, since it is balanced in all positions. Gate and globe valves may be converted into emergency valves by having the stems mechanically operated by electric motors, hydraulic pistons, and the like. Fig. 505 shows a section through a Crane hydraulically operated emergency gate valve.

Fig. 506 shows a partial section through an "Anderson triple-duty" emergency valve, and Fig. 507 a section through the pilot valve. A steam connection from the main line to the top of a copper diaphragm holds the pilot valve closed because of the large area above the diaphragm. A steam pipe connection from underneath the emergency piston of the triple-acting valve also leads to the pilot valve. In case a break occurs in the main steam line or branches, the pressure is removed from the top of the pilot valve, causing it to open, thus exhausting the pressure from beneath the emergency piston in the triple-acting valve. The boiler pressure on top of the emergency piston causes the valve to close. Pilot valves may be located at any desirable places, thus affording control from different points.

In the "Locke automatic engine stop system" the stop valve is operated by an electric motor which is controlled by contact points operated by a speed-limit device. (See Power, August, 1907, p. 471, for a detailed description.)

388. Check Valves. — Fig. 508, *A* to *D*, illustrates the different types of check valves in most common use. *A* is a *ball check*, *B* a *cup* or *disk*

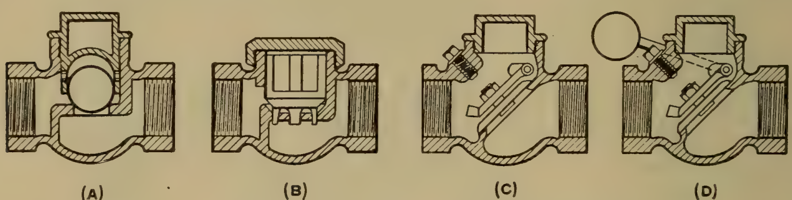


FIG. 508. Types of Check Valves.

check, *C* a *swing check*, and *D* a *weighted check*. Occasionally the valve body is fitted with a valve stem and handle for holding the disk against its seat, in which it is designated as a *stop check*. In *A* and *B* the valve seat is parallel to the direction of flow and the valve is held in place by its own weight and by the pressure of the fluid in case of reverse flow. In the swing check the seat is at an angle of about 45 degrees to the direction of flow. The latter construction is preferred as it offers less resistance to flow and there is less tendency for impurities to lodge on the valve seat. By extending the hinge of the swing through the body of the valve, a lever and weight may be attached as in *D* and the check will not open except at a pressure corresponding to the resistance of the weight. It thus acts as a relief valve and at the same time prevents a reversal of flow. *Stop checks* are usually inserted in boiler feed lines close to the boiler, and when locked, act as any ordinary stop valve and permit the piping to be dismantled or the regulating valve to be reground without lowering the pressure on the boiler. Since the wear

on check valves is excessive and necessitates frequent regrinding they are often mounted with *regrinding disks,* Fig. 508 (*C*), which may be "ground" against the seat without removing the valve from the line.

389. Blow-off Cocks and Valves. — The requirements of a good blow-off valve are that it shall furnish a free passage for scale and sediment, that it shall close tightly so as not to leak, and that it shall open easily without sticking or cutting. On account of the rather severe service to which such valves are subjected, they are made very heavy, with renewable wearing parts.

Fig. 509 gives a sectional view of a Crane ferrosteel valve. The bonnet is easily taken off and the disk removed to be refaced or replaced

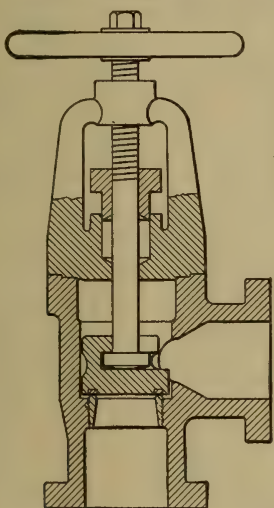


Fig. 509. Crane Ferrosteel Blow-off Valve.

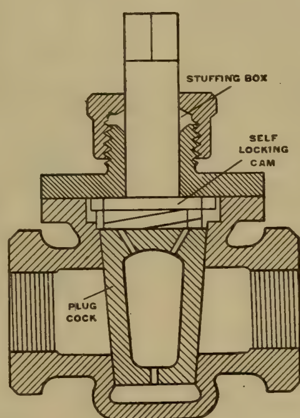


Fig. 510. A Typical Blow-off Cock.

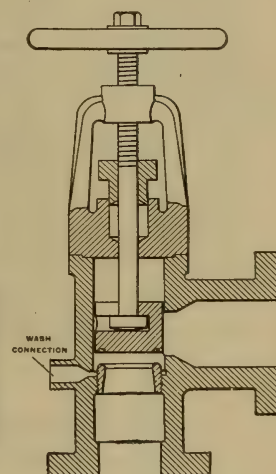


Fig. 511. Faber Blow-off Valve.

by a new one. The old disk is repaired by pouring in a hard Babbitt metal and facing it off flush. The seats are of brass and oval on top to prevent scale lodging between them and the disk, and are so made that they may be removed; but it has been found in practice that there is not much cutting of the seat, the damage usually being confined to the softer Babbitt metal which faces the disk.

Fig. 511 gives a sectional view of a Faber valve. When the disk, which makes a snug fit in the body of the valve, is in the position shown, the boiler discharge is practically shut off and any sediment lying on the seat is cleaned off by a jet of steam or water.

Fig. 510 shows a section through a typical *blow-off cock* of the straightway taper plug pattern with self-locking cam. Plug cocks are often used instead of valves on the blow-off piping.

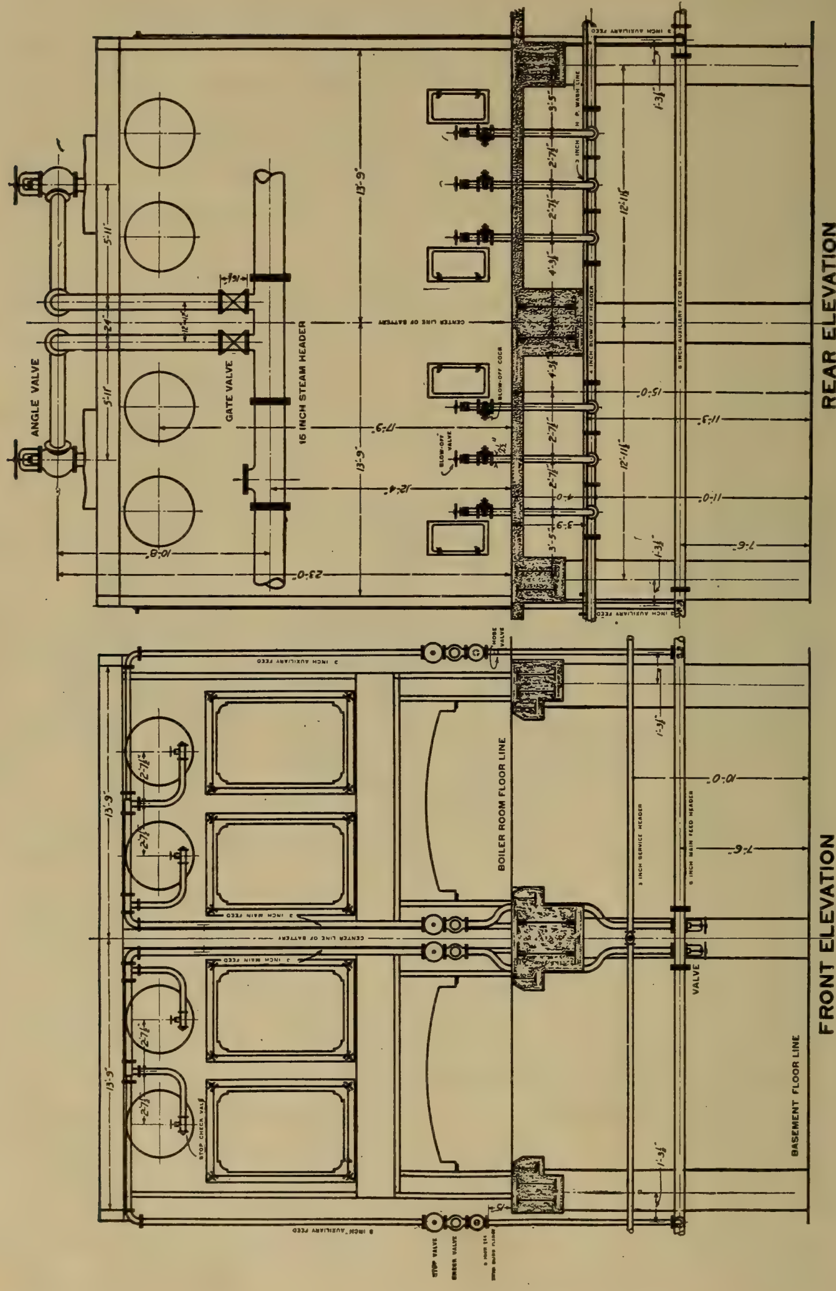


Fig. 512. Feed Water and Blow-off Piping, South Side Elevated Ry. Station, Chicago, Ill.

Current practice recommends the use of two valves, or rather one valve and one cock, in the blow-off line of each boiler. In most of the large stations a blow-off valve and a blow-off cock are installed as indicated in Fig. 512. The number and size of blow-off cocks are usually specified by city or state legislation. (For a description of various types of blow-off valves, see Power, Dec. 20, 1910, p. 2228.)

390. Safety Valves. — Fig. 513 shows a section through the simplest form of safety valve. The valve is held on its seat against the boiler pressure by a cast-iron weight as indicated. This type has the advantage of great simplicity, and can be least affected by tampering, since it requires so much weight that any additional amount which would seriously overload it can be quickly detected. For high pressure and large sizes of boiler this class of valve is entirely too cumbersome.

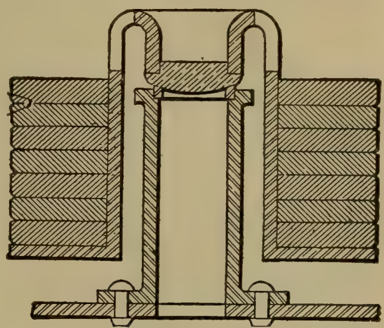


FIG. 513. "Dead-weight" Safety Valve.

Fig. 514 shows the general details of the common *lever safety valve.* The valve is held against its seat by a loaded lever, thereby enabling the use of a much smaller weight than the "dead-weight" type, since the resistance is multiplied by the ratio of the long arm of the lever

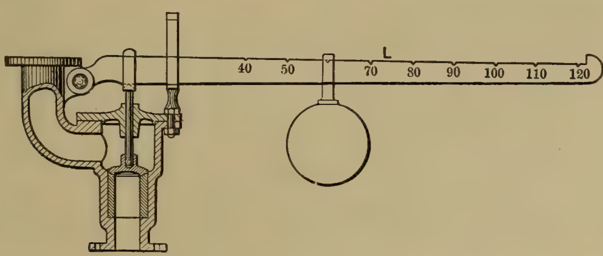


FIG. 514. Common Lever Safety Valve.

to the short one. The proper position of the weight is determined by simple proportion. Safety valves of the "dead-weight" or "lever" type are little used in modern practice, and their use is prohibited in U. S. marine service and in many states.

Fig. 515 shows a section through a typical *pop safety valve* in which the boiler pressure is resisted by a spring. This type of valve has practically supplanted all other forms. The boiler pressure acting upon the under side of valve V is resisted by the tension in spring S. As soon as the boiler pressure exceeds the resistance of the spring the valve

lifts from its seat and the steam escapes through opening O. The static pressure of the steam plus the force of its reaction in being deflected from the surface A holds the valve open until the pressure in the boiler drops about 5 pounds below that at which the valve is lifted. The additional area of valve exposed to pressure when the valve lifts causes it to open with a sudden motion which has given it its name, and it also closes suddenly when the pressure has fallen. These valves are arranged so that the spring tension may be varied without taking them apart, and provision is made for lifting the seats by means of a lever. The seats are of solid nickel in the best designs, to minimize corrosion.

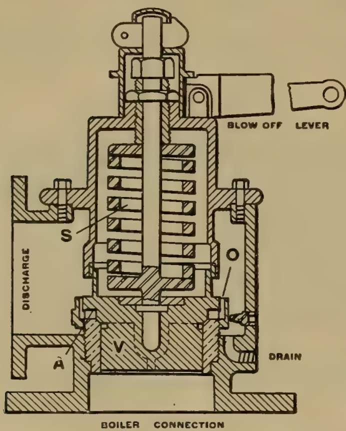


FIG. 515. Consolidated Pop Safety Valve.

The commercial rating of a safety valve is based upon the area exposed to pressure when the valve is closed.

The number and size of safety valves for a given boiler are ordinarily specified by city or state legislation.

The logical method for determining the size of safety valves is to make the actual opening at discharge sufficient to take care of all steam generated at maximum load, thus:

Let W = maximum weight of steam discharged, pounds per hour.
A = effective discharge area, square inches.
P = boiler pressure, pounds per square inch absolute.
L = lift of valve, inches.
α = angle of the valve seat with the horizontal.
K = coefficient determined by experiment.
D = diameter of valve, inches.

According to Napier's rule for the discharge of steam through unrestricted orifices

$$W = \frac{3600}{70} PA = 51.4\,PA. \tag{273}$$

Allowing for restriction of orifice

$$W = 51.4\,KPA. \tag{274}$$

Experiments by Philip G. Darling (Trans. A.S.M.E., Vol. 31, 1909, p. 123) gave a practically constant value of K = 0.925. Experiments conducted by the Consolidated Safety Valve Company gave the same

average value for K as determined by Darling. Substituting this value of K in (274),

$$W = 47.5\,PA. \tag{275}$$

For a flat-seated valve

$$A = \pi DL. \tag{276}$$

Whence

$$W = 149\,PDL \tag{277}$$

or

$$D = 0.0067\,\frac{W}{PL}. \tag{278}$$

For the almost universal 45-degree seated valve

$$A = \pi DL \text{ sine 45 degrees (approx.)}$$

$$= 0.707\,DL. \tag{279}$$

Whence

$$A = 105\,PDL \tag{280}$$

or

$$D = 0.0095\,\frac{W}{PL}. \tag{281}$$

The present rule of the United States Board of Supervising Inspectors is

$$a = 0.2074\,\frac{w}{P}, \tag{282}$$

in which

a = area of the safety valve in square inches per square foot of grate surface per hour.

w = pounds of water evaporated per square foot of grate surface per hour.

Formula (282) is derived by allowing a lift of $\frac{1}{32}$ of the nominal valve diameter and taking 75 per cent as the added restriction of a 45-degree over a flat seat, thus

$$a = \left(0.75\,\pi D \times \frac{D}{32}\right);$$

which, substituted in Napier's formula, gives $a = 0.2074\,\frac{w}{P}$.

The Consolidated Safety Valve Company's circular gives the following rated capacity of its nickel-seat pop safety valves:

TABLE 121.

RELIEVING CAPACITIES, CONSOLIDATED POP SAFETY VALVES, STATIONARY TYPE.

Pounds of Steam per Hour.

Size Valve, In.	Gauge Pressures (Lbs. per Sq. In.)														
	20	40	60	80	100	120	140	160	180	200	220	240	260	280	300
$1\frac{1}{2}$	600	950	1300	1650	2000	2,340	2,690	3,030	3,380	3,720	4,070	4,410	4,760	5,110	5,470
2	880	1390	1890	2400	2900	3,400	3,900	4,410	4,910	5,420	5,920	6,430	6,930	7,430	7,940
$2\frac{1}{2}$	1100	1730	2360	3000	3620	4,250	4,880	5,500	6,140	6,760	7,400	8,030	8,650	9,300	9,900
3	1430	2250	3070	3890	4700	5,530	6,350	7,170	8,000	8,800	9,620	10,400	11,200	12,100	12,900
$3\frac{1}{2}$	1810	2830	3860	4880	5910	6,950	7,960	9,020	10,000	11,100	12,100	13,100	14,200	15,200	16,300
4	2060	3240	4410	5580	6770	7,950	9,120	10,300	11,500	12,600	13,800	15,000	16,200	17,300	18,500
$4\frac{1}{2}$	2480	3900	5310	6730	8150	9,570	11,000	12,400	13,800	15,200	16,700	18,100	19,500	20,900	22,400
5	2940	4620	6300	7970	9650	11,330	13,000	14,700	16,400	18,100	19,700	21,400	23,100	24,800	26,500

391. Back-pressure and Atmospheric Relief Valves. — These valves are for the purpose of preventing excessive back pressure in exhaust pipes. In non-condensing plants such valves are designated as *back-pressure valves* and in condensing plants as *atmospheric relief valves.* In the former the valve is usually adjusted so that a pressure of one to five pounds above the atmosphere is necessary to lift it from its seat; in the latter the valve lifts at about atmospheric pressure. They are practically identical in construction, differing only in minor details. A slight leakage in the back-pressure valve is of small consequence, but in an atmospheric relief valve it may seriously affect the degree of vacuum and throw unnecessary work upon the air pump, hence it is customary to "water-seal" the latter. Fig. 516 shows a section

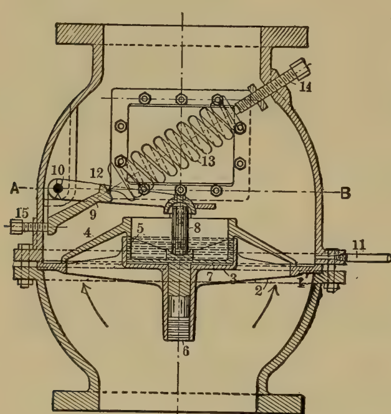


FIG. 516. Foster Back-pressure Valve.

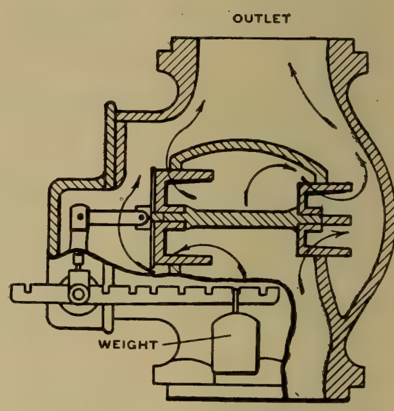


FIG. 517. Davis Back-pressure Valve.

through a typical back-pressure valve. The valve proper consists of a single disk moving vertically. The valve stem is in the form of a piston or dashpot which prevents sudden closing or hammering. The pressure holding the valve against its seat is regulated by a spring. When the back pressure becomes greater than atmospheric plus that added by the spring, the valve raises from its seat and relieves it.

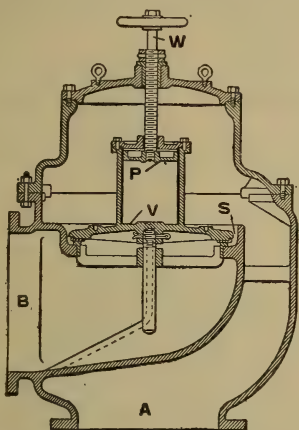
Fig. 517 shows a section through a Davis back-pressure valve, in which the resisting pressure is varied by means of a lever and weight.

Fig. 486 shows the application of a back-pressure valve to a typical heating system.

Fig. 518 shows a section through a typical *atmospheric relief valve.* Opening B is connected to the exhaust pipe and opening A leads to the atmosphere. Under normal conditions of operation atmospheric pressure holds valve V against its seat. Water in groove S "water-seals" the seat and prevents air from being drawn into the condenser.

In case the pressure in pipe B becomes greater than atmospheric it lifts valve V from its seat and is relieved. Piston P acts as a dashpot and prevents the valve from slamming.

Fig. 519 shows a section through an atmospheric relief valve in which the weight of the valve is counterbalanced or even overbalanced by an adjustable weight and lever, thereby permitting the valve to open at or below atmospheric pressure, as may be desired.



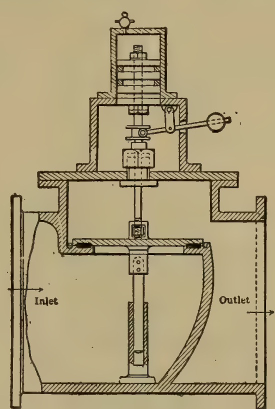


FIG. 518. Crane Atmospheric Relief Valve. FIG. 519. Acton Atmospheric Relief Valve.

392. Reducing Valves. — It is often necessary to provide steam at different pressures in the same plant, as in the case of a combined power and heating plant. To effect this result the reduction in pressure is accomplished by passing the steam through a *reducing valve,* which is but an automatically operated throttle valve. There are many different forms, the operation of all being based upon the same general principles.

In the Kieley valve, Fig. 520, the low-pressure steam acts upon the top of flexible diaphragm D, and the weighted lever L (which may be adjusted to give the desired reduction in pressure) acts upon the other side. The movement of the diaphragm causes the balanced valve V at the upper end of the spindle to open or close, as may be necessary to maintain the desired lower pressure. Inertia weights T and C prevent chattering.

Fig. 521 shows a section through a class G Foster *pressure regulator* or reducing valve. In operation, steam enters at A and passes through the main valve port H to the outlet B. Steam at initial pressure passes through port C to chamber P and thence to the top of piston T through port L, opening the main valve U. Steam at delivery pressure passes through E and raises the diaphragm V against the pressure of spring R, allowing spring W to close the auxiliary valve X. The pressure in

chamber J is then equalized by the reduced pressure in ports G and the under side of piston X, and thus allows spring Y to close the main valve which is then held to its seat by the initial pressure. Any reduction in delivery pressure is transmitted to diaphragm V, and permits spring to open auxiliary valve X, thereby admitting steam to the top of piston T, as previously explained. The delivery pressure is adjusted by screw D; thus increasing the tension of spring R increases the discharge pressure; and *vice versa.* The adjustment once made, the delivery pressure will

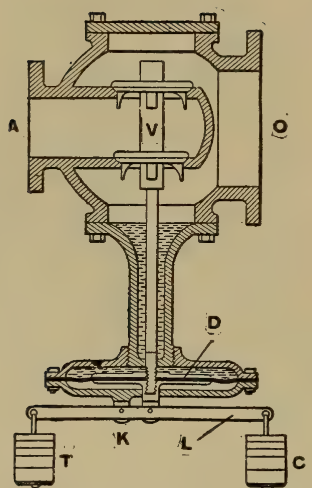


FIG. 520. Kieley Reducing Valve.

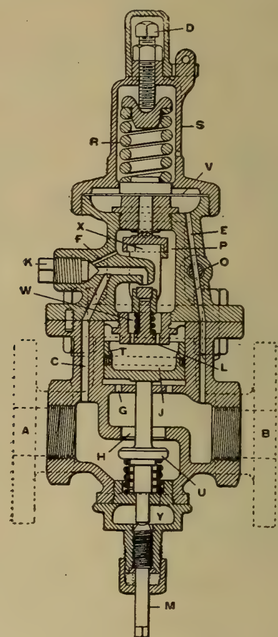


FIG. 521. Foster Pressure Regulator.

remain constant, regardless of any variable volume of discharge or of the initial pressure, so long as the latter is in excess of the delivery pressure. W, Fig. 489, shows the application of a reducing valve to an exhaust steam heating system. Live steam is led to the valve through pipe A. It will be noted that the pipe leading from the valve to the heating system is much larger than the high-pressure supply pipe on account of the increase in volume of the low-pressure steam. Reducing valves should always be by-passed to permit of repairs without shutting down the system. Care should be taken in not selecting too large a reducing valve, as the valve lift is very small and the larger the valve the less will be the lift for a given weight of flow and consequently the greater the wire drawing and erosion of the valve seat.

393. Foot Valves. — Whenever a long column of water is to be moved in either suction or delivery pipe it is customary to place a check valve near the lower end of the column to prevent the water from backing up when the pump reverses or shuts down. The check valve placed at the end of the suction pipe is called a *foot valve.* Any check valve may be used as a foot valve, though practice limits the choice to the disk

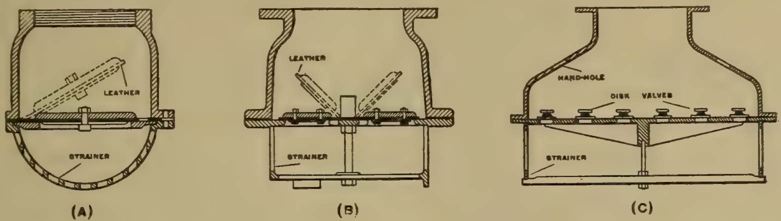


FIG. 522. Types of Foot Valves.

or flap type as illustrated in Fig. 522. To prevent rubbish from destroying the action, a strainer or screen is generally incorporated with the body of the valve. *A*, Fig. 522, illustrates a *single-flap*, *B* a *multi-flap*. and *C* a *disk* valve composed of a nest of small rubber valves. The single-flap are usually made in sizes $\frac{3}{4}$ to 6 inches, the multi-flap 7 to 16 inches, and the disk valve in all commercial sizes from $\frac{3}{4}$ to 36 inches. For large sizes, 16 to 36 inches, the multi-disk valve is given preference, since a number of the disks may be disabled without destroying its operation.

The Use and Abuse of Globe Valves: Power and Engr., Jan., 1909, p. 10.
Gate Valves in Steam Pipe Lines: Power and Engr., Feb. 16, 1909, p. 320.
Types of Check Valves and Their Operation: Power and Engr., July 6, 1909, p. 11.

CHAPTER XVI.

LUBRICANTS AND LUBRICATION.

394. General. — The losses due to the friction of the working parts of machinery include considerably more than the mere loss of power, namely, the depreciation resulting from wear of bearings, guides, and other rubbing surfaces, and the expense arising from accidents traceable to excessive friction. The power absorbed in overcoming friction varies with the type of plant and the character of machinery and is seldom less than 5 per cent and often greater than 30 per cent of the total power developed. In large central stations these losses approximate 8 per cent and in weaving and spinning mills will average as high as 25 per cent. (Trans. A.S.M.E., 6–465.) These figures refer to properly lubricated plants operating under normal conditions. The proper selection of lubricant is therefore a very important problem, since, besides the cost of the lubricant itself, the loss in power and in wear and tear to machinery is no small item. A change of lubricant may frequently result in marked increase in economy of operation. Table 122 gives an idea of the saving effected in power by the proper selection of lubricants in a number of mills. (Power, May 12, 1908, p. 752.) The net financial gain depends, of course, upon the cost of the oil. As a general rule a 10 per cent reduction in friction horse power will more than equal the cost of lubricants for one year. The lubricants most commonly met with in power plant practice are conveniently classified as oils, greases, and solids, and are of animal, mineral, or vegetable origin.

Reference books: Archbutt and Deeley, Lubrication and Lubricants; Redwood Lubricants; W. M. Davis, Friction and Lubrication; Gill, Oil Analysis; Robinson, Gas and Petroleum Engines; Thurston, Friction and Lost Work; Gill, Engine Room Chemistry.

395. Vegetable Oils. — Except for certain special purposes and for compounding with mineral oils these possess lubricating properties of little practical value, since they decompose at comparatively low temperatures and have a tendency to become thick and gummy. The vegetable oils sometimes employed are linseed, cottonseed, rape, and castor.

396. Animal Fats. — Many animal fats have greater lubricating power than pure mineral oils of corresponding viscosity but are objec-

tionable on account of their unstable chemical composition. They decompose easily, especially in the presence of heat, and set free acids which attack metals. They are seldom used in the pure state and are usually compounded with mineral oils. The animal products used in this connection are tallow, neat's-foot oil, lard, sperm, wool grease, and fish oil, the first named being the most important. In cylinder lubrication, especially in the presence of moisture, the addition of 2 to

TABLE 122.

EXAMPLES OF REDUCTION IN FRICTION DUE TO PROPER SELECTION OF LUBRICANTS.

No. of Test.	Country.	Plant.	Mill Oils. Test I.		New Oils. Test II.		Per Cent of Transmission to Full Load.		Power Reductions.	
			Full Load, I.H.P.	Transmission, I.H.P.	Full Load, I.H.P.	Transmission, I.H.P.	Test I.	Test II.	Full Load, Per Cent.	Transmission, Per Cent.
1	America	Cotton	543.21	192.70	481.75	168.90	35.4	35.0	11.31	12.35
2 A	America	Worsted	611.60	...	596.30	...	...	...	2.50	...
B	America	Worsted	702.90	...	648.70	...	...	...	7.80	...
3	America	Cotton	786.00	...	758.00	...	...	...	3.56	...
4 A	England	Cotton	1408.60	356.00	1301.80	319.30	25.3	24.5	7.60	10.30
B	England	Cotton	1428.40	357.90	1358.70	348.90	25.0	25.7	4.90	2.50*
5	England	Worsted	348.10	111.10	327.50	99.50	31.9	30.4	5.90	10.40
6	England	Weaving	495.00	146.60	453.60	127.50	29.6	28.1	8.40	13.00
7	Ireland	Linen	110.70	49.90	93.10	38.60	45.0	41.4	15.90	22.70
8 A	Scotland	Woolen	177.70	61.80	164.60	56.10	34.7	34.0	7.40	9.20
B	Scotland	Woolen	325.10	161.40	293.50	147.30	49.6	50.2	9.70	8.70
9	Germany	Cotton	263.41	114.03	239.35	97.11	43.2	40.5	9.10	14.80
10 A	Germany	Worsted	341.36	118.24	290.53	95.67	31.7	32.9	14.90	19.10
B	Germany	Worsted	341.36	141.29	299.30	119.28	41.3	39.8	12.30	15.57‡
11	Germany	Jute	1135.20	362.60	1034.20	328.10	31.9	31.7	8.89	9.51
12	Russia	Cotton	1238.80	...	1069.10	...	...	...	13.70	...
13	India	Cotton	642.60	230.70	596.80	202.20	35.9	33.9	7.10	12.40
14	Japan	Cotton	346.60	...	313.60	...	...	...	9.50	...
15	India	Flour	364.70	...	336.80	...	...	...	7.70	...
16	England	Paper	465.40	...	390.40	...	...	...	16.20	...
17	Germany	Paper	511.37	...	482.43	...	...	...	5.60	...
18	England	Brass shop	6.74x	1.77x	5.12x	1.53x	26.2	29.8	24.00	13.80
19	England	Iron shop	137.80	74.90	116.00	68.10	54.3	58.7	15.80	9.10
20	England	Wood shop	84.00	31.60	65.30	25.40	37.6	38.8	22.30	19.60

* Same oil after nine months' use. † Not full load of mill. ‡ Morning load.
x = Electrical units.

5 per cent of acidless tallow seems to make the oil adhere better to the metal surfaces and increases the lubricating effect, while the proportion is so small that ill effects from corrosion or gumming are scarcely perceptible.

397. Mineral Oils. — These are all products of crude petroleum and form by far the greater part of all lubricants. They present a wider range of lubricating properties than those derived from animal or vegetable sources, the thinnest being more fluid than sperm and the thickest more viscous than fats and tallows. They are not easily oxidized, do not decompose, become rancid, or contain acids.

Crude American petroleum of specific gravity 0.802 may yield the following commercial products. ("Gas and Petroleum Engines," W. Robinson.)

TABLE 123.

		Average Percentage.	Specific Gravity.	Boiling Point, Degrees F.
Light Oils.				
Petroleum ether	Cymogene	traces	0.590	32
	Rhigolene	0.1	.625–.631	64
	Gasoline	1–1.5	.635–.658	85–155
Petroleum spirit	C. Naphtha	10	.680–.700	140–212
	B. Naphtha	2–2.5	.717–.72	175–245
	A. Naphtha (benz.)	2–2.5	.742–.745	212–265
Burning oils, kerosene.	Water white	12–20	.780–.785	300–570
	Ordinary kerosene	40–55	.800–.810	300–680 and upwards
Fuel oils	For making oil gas; fuel	....	0.85	
Heavy oils	Lubricating oils	17.5	.885–.920	480 and upwards
	Paraffin wax	2	.908 at 60 deg. F.	
	Residium	5–10		

Mineral lubrication oils may be classified as

(1) *Distilled oils,* which are produced by distillation from crude petroleum and made pale, amber colored, and transparent by treatment with acid and alkali.

(2) *Natural oils,* which are prepared from crude petroleum, from which grit, suspended and tarry impurities have been removed. They are dark and opaque and are rich in lubricating properties.

(3) *Reduced oils,* or heavy natural oils, from which the lighter hydrocarbons have been evaporated and from which the tarry residue has been removed by filtration.

398. Solid Lubricants. — Dry graphite, soapstone, and mica are sometimes used as lubricants, though they are usually mixed with grease or oils. They cannot easily be squeezed or scraped from between the surfaces, and are consequently suitable where very great weights have to be carried on small areas and when the speed of rubbing is not high. The coefficient of friction of such lubricants is high, and when economy of power is essential better results may be secured by the use of liberally proportioned rubbing surfaces and liquid lubricants. Under

certain conditions of pressure and speed these lubricants will sustain, without injury to the surfaces, pressures under which no liquid would work.

Deflocculated graphite suspended in oil or water, and designated commercially as "oildag" and "aquadag" respectively, is finding favor with many engineers. Graphite in this deflocculated condition remains suspended indefinitely in water and oil, readily coheres to the journal, has great wearing properties, and is easily applied to the wearing surfaces. From numerous and long-continued trials it appears that 0.35 per cent serves adequately for all purposes. Temperature curves of deflocculated graphite in combination with various carrying fluids are given in Fig. 523. For further data pertaining to the curves in Fig. 523 and for an extensive discussion on the subject of lubrication consult *Lubrication and Lubricating*, by C. F. Maberg, Jour. A.S.M.E., Feb. and May, 1910.

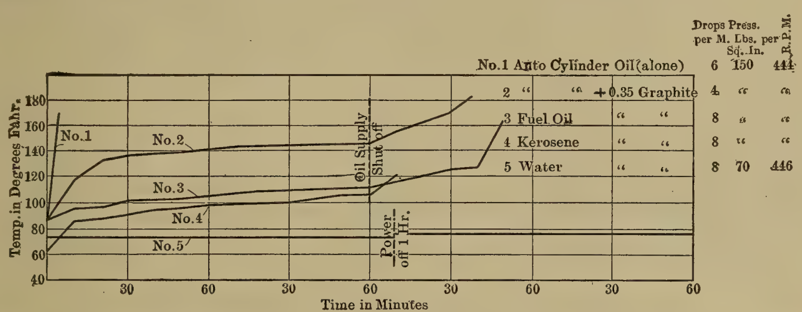


FIG. 523. Tests of Graphite Mixed with Various Lubricants.

399. Greases. — Under this name may be included the various compounds which consist of oils and fats thickened with sufficient soap to form, at ordinary temperatures, a more or less solid grease. Those usually employed are lime, soda, or lead soaps, made with various fats and oils. "Engine" greases are thickened with a soap made from tallow or lard oil and caustic soda, and often contain neat's-foot oil, beeswax, and the like. For exceptionally heavy pressures, graphite, soapstone, and mica are sometimes added to the grease. Table 124 gives an idea of the characteristics of a number of greases. (Prac. Engineer, U. S., Apr., 1911, p. 293.) The friction tests were made on a small Thurston oil testing machine, 320 r.p.m. and bearing pressure of 240 pounds per square inch of projected area. These results are purely comparative under the given conditions of rubbing surfaces, speed and pressure. For results of these greases tested on a large Olsen oil machine consult reference given above.

Commercial Lubricating Greases: Prac. Engineer, U. S., Apr., 1911, p. 293; *Tests of Grease Lubrication*, Ibid., p. 295; Am. Mach., Aug. 24, 1911, p. 356; Power, Nov. 8, 1910, p. 1998.

TABLE 124.

Type.	Class.	Melting Point, Deg. F.	Per Cent Soap.	Kind of Soap.	Per Cent Free Acid as Oleic.	Average Coefficient Friction.
A Mineral	Summer	167	38	Lime	Trace	0.075
B Mineral	Summer	178	20	Lime	0.3	0.054
C Mineral	Winter	165	23	Lime	6.1	0.063
D Mineral	Winter	163	16	Lime	0	0.057
E Mineral	Winter	142	19	Lime	Trace	0.046
F Tallow No. 3	Winter	125	1.4	Potash	0	0.022
G Tallow No. XX	Summer	120	2.1	Potash	0	0.029
H Lard oil	....	41	0	....	....	0.011

Type.	Final Coefficient Friction After 3-Hr. Run.	Maximum Temperature of Bearing Above that of Room, Degs. F.	Final Temperature of Bearing Above that of Room at End of 3-Hr. Run, Degs. F.
A Mineral	0.075	70	68
B Mineral	0.050	70	58
C Mineral	0.063	76	65
D Mineral	0.054	69	58
E Mineral	0.046	58	50
F Tallow No. 3	0.012	38	18
G Tallow No. XX	0.018	45	32
H Lard oil	0.010	13	12

400. Qualifications of Good Lubricants. — A good lubricant should possess the following qualities:

(1) Sufficient "body" to prevent the surfaces from coming into contact under conditions of maximum pressure.

(2) Capacity for absorbing and carrying away heat.

(3) Low coefficient of friction.

(4) Maximum fluidity consistent with the "body" required.

(5) Freedom from any tendency to oxidize or gum.

(6) A high "flash point" or temperature of vaporization and a low congealing or "freezing point."

(7) Freedom from corrosive acids of either metallic or animal origin.

Lubricating oils are identified by certain tests which are used by refiners in grading and classifying the oils and by consumers in buying them. These tests usually cover the following:

(1) Identification of the oil, whether a simple mineral, animal or vegetable oil or a mixture.

(2) Density or gravity.

(3) Viscosity.

(4) Flash point.

(5) Burning point, fire test.
(6) Acidity.
(7) Coefficient of friction.
(8) Cold test.

401. Identification of Oil. — The chemical analysis of oils lies in the province of the chemist, but some of the characteristics may be readily determined by a few simple tests. To detect admixtures of fatty oils in mineral oil a small quantity is heated in a test tube for 15 minutes with small pieces of either metallic sodium or caustic potash. If fatty oil is present, saponification takes place and the soap formed will rise to the top as a semi-solid mass and the amount may be estimated. Tarry matter may be detected by dissolving a small quantity of oil in from 10 to 20 times its bulk of gasoline; the tar and other insoluble matter will separate and collect at the bottom.

For a number of single tests for identifying the various constituents of compound or adulterated oils consult "Engineering-Room Chemistry," by Augustus H. Gill.

402. Gravity. — The density or specific gravity is conveniently determined by means of a hydrometer, which, in the oil trade, is graduated according to the Baumé scale. The relationship between specific gravity and degrees Baumé at a temperature of 60 degrees F. may be expressed:

$$\text{Specific gravity} = \frac{140}{130 + \text{degrees Baumé}}. \tag{283}$$

Table 125 gives the specific gravity and gravity Baumé of a number of lubricating oils.

TABLE 125.

SPECIFIC GRAVITY AND GRAVITY BAUMÉ OF A NUMBER OF LUBRICANTS.

	Specific Gravity.	Gravity Baumé.	Flash Test, Degrees F.
Water	1.000	10	....
Cylinder oil	.9090	24.5	575
Cylinder oil	.8974	26	540
Heavy engine oil	.9032	25.5	411
Medium engine oil	.9090	24	382
Light engine oil	.8917	27	342
Castor machine oil	.8919	27	324
Lard oil	.9175	23	505
Sperm oil	.8815	29	478
Tallow oil	.9080	24.5	540
Cottonseed oil	.9210	22	518
Linseed oil	.9299	19	505
Castor oil (pure)	.9639	15	....
Palm oil	.9046	25	405
Rape-seed oil	.9155	23	....
Spindle oil	.8588	33	312

403. Viscosity. — Viscosity may be defined as the degree of fluidity or internal friction of an oil. It is sometimes called the "body." It is determined by a viscosimeter. There are a number of different instruments for this purpose but no recognized standard instrument or method, so that "viscosity" conveys no meaning unless the name of the instrument, the temperature, and the amount of oil tested are given. Nearly all instruments are of the orifice type; that is, the viscosity of an oil is taken as the number of seconds required for a given amount, usually 50 cubic centimeters, to flow through an orifice at a given temperature. By "specific viscosity" is meant the ratio of the time required for the oil to run out to that of an equal quantity of water at 60 degrees F. The viscosity of engine oils is usually taken at 70 degrees F. and of cylinder oils at 212 degrees F.

404. Flash Point. — The flash point is determined by heating a sample of oil in an open or closed cup at the rate of 15 degrees F. per minute until a spark will ignite the vapor. The temperature at which this occurs is the flash point. So much depends upon the extent of oil surface exposed, size of spark, distance spark is held from the oil at the time of ignition, and the dimensions of the cup, that there may be considerable variation in the flash point as obtained by different experimenters.

405. Burning Point, or Fire Test. — By continuing the application of heat and noting the temperature at which the oil takes fire and continues to burn, the burning point is obtained. The higher the temperature under which the oil must work the higher the fire test required, so that it will not decompose or volatilize. Too high a fire test gives an oil that does not atomize readily enough to reach all parts of the cylinder.

406. Acidity. — The presence of free acid is determined by shaking up equal quantities of oil and water and testing with litmus paper. Another simple test is as follows: A small quantity of oil is placed in a test tube with a little cupric oxide (Cu_2O) and subjected to a gentle heat for three or four hours. The reaction with the copper turns the solution green if fatty acid is present and blue if vegetable acid is present.

407. Cold Test. — The "cold test" is the temperature at which the oil will just flow. The sample is solidified by means of a freezing mixture and the temperature noted when it softens sufficiently to flow.

408. Friction Test. — The coefficient of friction as determined from friction-testing machines is useful in obtaining a comparison of oils under the test conditions, but gives little information concerning the action of the oil under the widely different conditions found in actual practice. Table 126 gives the physical properties of a number of lubricating oils, with their particular zone of application.

409. Atmospheric Surface Lubrication. — In a general sense all journals, slides, and "atmospheric" surfaces should be lubricated with straight mineral oils (as free from paraffin as possible), except when in contact with considerable water, in which case it is advisable to add 20 to 30 per cent of lard oil. Vegetable oils, paraffin oils, and animal oils (except lard oil as above stated) are not recommended for general engine and dynamo service. The test requirements of a number of classes of lubricants are outlined in Table 89 and represent current practice. Bearings, guides, and all external rubbing surfaces may be lubricated in a number of ways. (1) They may be given an *intermittent* application of oil, as, for example, with an oil can; (2) they may be equipped with oil cups with *restricted* rates of feed; and (3) they may be *flooded* with oil. The relative lubricating values of the systems have been estimated approximately as follows (Power, December, 1905) p. 750):

	Coefficient of Friction.	Comparative Value.
Intermittent	0.01 and greater	72 and less
Restricted feed	0.01 to 0.012	79 to 86
Flooded bearing	0.00109	100

410. Intermittent Feed. — Intermittent applications are ordinarily limited to small journals, pins, and guides which are subject to light pressures and which do not easily permit of oil or grease cups, as, for example, parts of the valve gear of a Corliss engine, governors, and link work. On account of the labor attached and the frequent doubt about the oil reaching the wearing surfaces this method of lubrication is limited as much as possible even in the smallest plants.

411. Restricted Feed. — In the average power plant the major part of the lubrication is effected by means of oil cups which are filled at intervals by hand or by mechanical means, the oil being fed from the cup by drops, according to the requirements.

412. Oil Bath. — In large power plants the principal journals and wearing parts are supplied with a continuous flow of oil which completely "floods" the rubbing surfaces. The oil is forced to the various parts either by gravity from an elevated tank or by pressure from a pump. After the oil leaves the bearings it flows into collecting pans, thence into a receiving and filtering tank, and finally is pumped back into an elevated reservoir and used over and over again. The little lost by leakage and depreciation is replenished by the addition of new oil to the system.

TABLE 126.

PHYSICAL CHARACTERISTICS OF A NUMBER OF LUBRICANTS.

(*Power*, December, 1905, p. 750.)

Kind of Oil.	Use and Adaptation.	Gravity, Degrees.	Cold Test, Degrees.	Flash Test, Degrees.	Fire Test, Degrees.	Viscosity at 70 Degrees.
High-pressure cylinder oil.	For steam cylinders using dry steam at pressures from 110 to 210 pounds.	25 to 24.5	30	600 to 610	645 to 660	175 to 205
General cylinder oil . .	For steam cylinders using dry steam at 75 to 100 pounds. For air compressor cylinders when made from steam-refined mineral stock and when viscosity is 200.	26 to 25.5	30	550 to 585	600 to 630	180 to 190
Wet cylinder oil. (Remark 1.)	For use where the steam is moist, especially in compound and triple expansion engines.	25.8 to 25.3	30	560 to 585	600 to 630	150 to 185
Gas engine cylinder oil. (Remark 2.)	For gas engine cylinders. Neutral mineral oil compounded with an insoluble soap to give body.	26.5	30	320	350	300
Automobile gas engine oil. (Remark 3.)	For automobile gas engines and similar work.	29.5	30	430	485	195
Heavy engine and machinery oils.	For heavy slides and bearings, shafting, and horizontal surfaces.	30.5 to 29.5	30	400	440 to 450	170 to 195
General engine and machine oils.	For high-speed dynamos and machines.	30.8 to 30	30	400 to 420	450 to 470	175 to 190
Fine and light machine oils.	For fine work, from printing presses to sewing machines and typewriter oils. With a cold test of 25° to 28° and a viscosity of 140° this makes an excellent spindle oil.	32.5 to 30.2	30	400	440	110 to 160
Cutting and heat dissipating oils. (Remark 4.)	For cutting tools, screw cutting and similar work.	27 to 23	30	410 to 420	475 to 480	210 to 175
Refrigerating oils.	For ice machinery.	30.2	0	200	225	165
Wet service and marine oils. (Remark 4.)	For marine service, or where a great deal of moisture must be handled.	28	30	430	475	230
Greases.	They are used in special work and for heavy pressures moving at slow velocities.					

Remark 1. — May contain not over 2 to 6 per cent of refined acidless tallow oil in the high-pressure oils and not over 6 to 12 per cent in the low-pressure oils.

Remark 2. — The reason for using an insoluble soap such as oleate of aluminum is that it is impossible to decompose the soap with a high heat; the soap, although not a lubricant, is a vehicle for carrying some oil.

Remark 3. — Owing to a lack of body, this oil will not interfere with the sparking by depositing carbon on the platinum point.

Remark 4. — May contain 30 to 40 per cent of pure strained lard oil.

413. Oil Cups. — Fig. 524 illustrates the application of *sight-feed* oil cups to the crosshead and slides of a reciprocating engine. The oil is fed into the cups by hand and gravitates to the rubbing surfaces, the rate of flow being regulated by a needle valve. Cups *A* and *B* feed directly to the crosshead guides, but the oil from cup *D* flows to the bottom orifice *O*, from which it is wiped by a metallic wick *S*, and carried by gravity to the wrist pin.

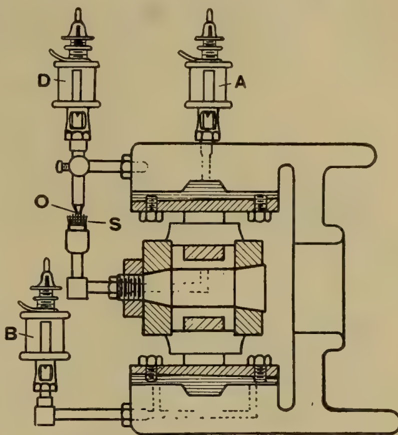


Fig. 524. Oil-cup Lubrication, Hand-filled.

414. Telescopic Oiler. — Fig. 525 shows the application of a *telescopic oiler* to a crosshead and guides. *O* and *C* are sight-feed oil cups, the former feeding directly to the top guide through the tube *S*. The oil from *C* flows by gravity through the swing joint into the telescopic tubes *P*, *R*, and thence to the pin through the lower swing joint as indicated.

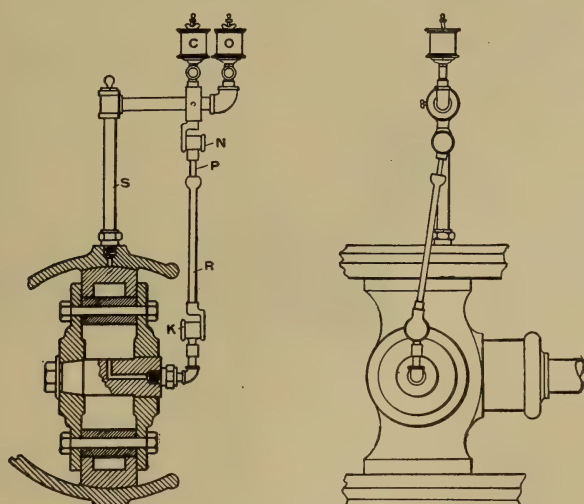


Fig. 525. Nugent's Telescopic Oiler.

As the crosshead moves back and forth, the pipe *P* slides into and out of pipe *R*, the oil being thus conducted directly to the pin without wasting. A device of this type installed on a high-speed automatic engine at the

Armour Institute of Technology has been in operation for three years without cost for repair or renewal.

415. Ring Oiler. — Small high-speed engines are often oiled by the *oil-ring* system, as illustrated in Fig. 526. The shaft is encircled by

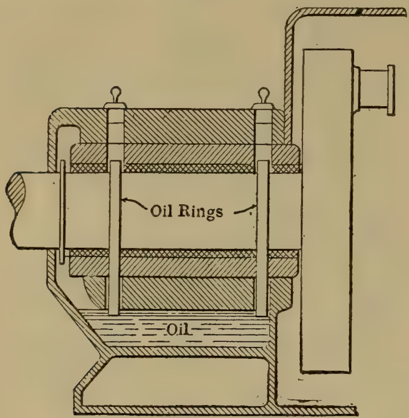


Fig. 526. Oil-ring Lubrication.

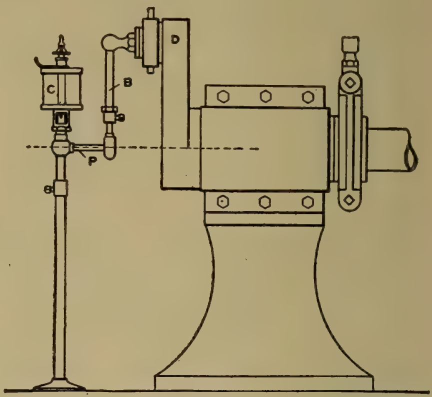


Fig. 527. Centrifugal Oiler.

several loose rings which dip into a bath of oil in the base of the pedestal or frame and, rolling on the shaft as it turns, carry oil to the top of the shaft where it spreads to the bearings. In some cases the rings are replaced by loops of chain.

Ring Lubrication: Zeit. d. Ver. Deutscher Ing., Aug. 10, 1907.

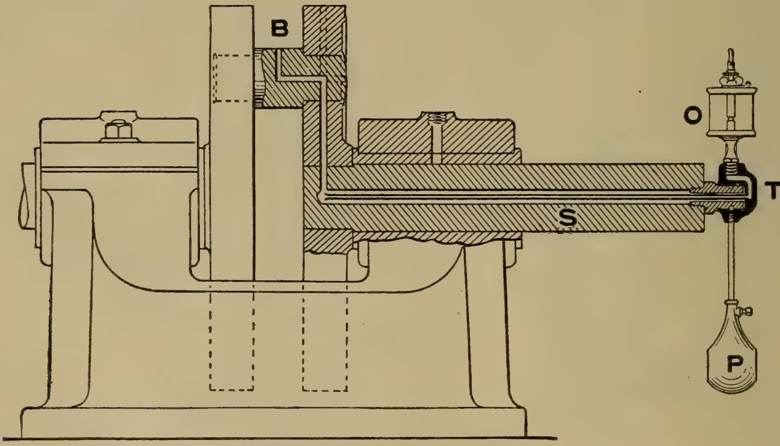


Fig. 528. Pendulum Oiler.

416. Centrifugal Oiler. — Fig. 527 illustrates the application of a *centrifugal oiler* to a side-crank engine. The oil supply is regulated by the sight-feed cup C and flows by gravity to the pipe P in line with

the center of the crank shaft. Centrifugal force throws the oil outward through pipe B to the center of the pin D, which is drilled longitudinally and radially so as to distribute the oil upon the bearing surface.

417. Pendulum Oiler. — Fig. 528 illustrates the application of a *pendulum oiler* to the crank pin of a center-crank engine. Oil cups and pendulum P are fastened to the crank shaft S by trunnion T. The pendulum holds the cup vertical, since the friction of the trunnion is not sufficient to revolve it. Oil flows along the center of the crank shaft under the head of oil in cup O and is thrown outward to bearing B by centrifugal force.

418. "Splash Oiling." — In some high-speed engines the crank, connecting rod, and crossheads are inclosed by a casing, the bottom of which is filled with oil to such a depth that at each revolution of the crank, the end of the connecting rod is partly submerged. The result is that the oil is splashed into every part of the chamber, and the crank pin, crosshead pin, and crosshead slides practically run in an oil bath.

419. Gravity Oil Feed. — Fig. 529 illustrates a simple *gravity oil-feed* system. The oil to the engine is supplied from the oil tank by pipe D under pressure corresponding to the height of the tank above the oil cups. After performing its function the oil gravitates to the filter and from the latter to the oil reservoir, from which it is pumped back to the supply tank, the overflow being returned to the reservoir through pipe N. Operation is interrupted only when new oil is to be added to the system from the barrel through the flexible filling pipe. In case the oil tank is put out of commission, or the supply pipe becomes clogged, full pump pressure may be used by closing valves R and S and opening valve E. The make-up oil is small in amount compared to the quantity circulated. The reclaiming and purifying of the oil are essential if the bearings are to be flooded, otherwise the cost of oil would be prohibitive. At the power house of the South Side Elevated Railway the daily circulation (24 hours) of engine oil is approximately 1500 gallons. The make-up oil amounts to eight gallons.

An objection sometimes made to the above system is that the varying heights of oil in the supply tank may cause considerable variation in pressure at the oil cups, causing them to feed faster when the tank is full and slower when the tank is nearly empty. This applies only to installations where the supply tank is filled intermittently.

420. Low-pressure Gravity Feed. — Fig. 530 shows the application of a low-pressure oiling system in which the level in the sight feeds is kept constant. A is the main supply tank, B^1 and B^2 the upper and lower gauges indicating the oil level, C the supply pipe running to the engines, and D a small standpipe closed at one end and vented near the

top. The reservoir is supplied with oil by the valve marked "inlet." When the tank is filled the oil rises in the standpipe *D* a corresponding height. The inlet valve is then closed and the oil in the standpipe feeds down to the level of the sight feeds or to a point where the air will enter the bottom of the tank. This will be the constant oil level, since oil flows from the tank only in proportion to the amount of air

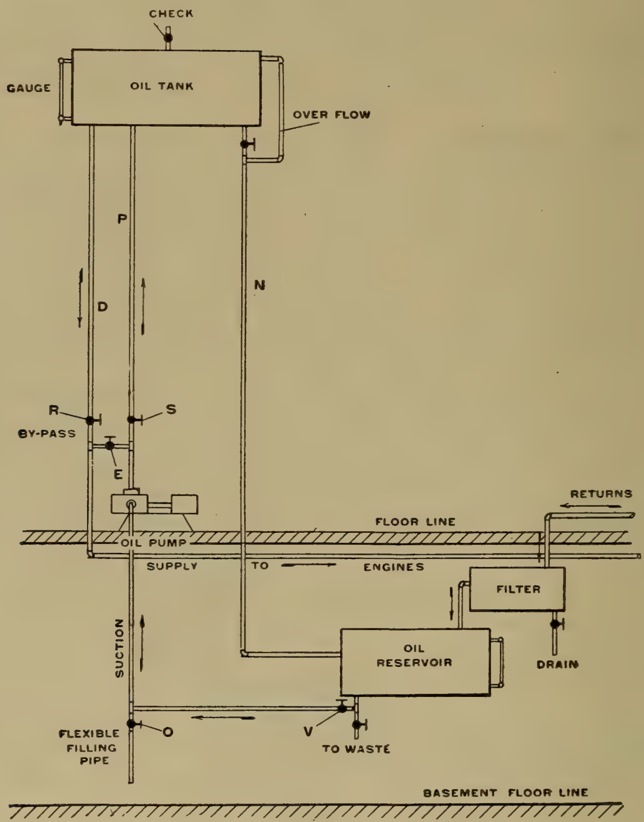


Fig. 529. Simple Gravity Feed System.

admitted. A head of 6 inches has been found to give the best results. (Engineer, U. S., March 16, 1903, p. 243.)

421. Compressed-air Feed. — Fig. 531 shows diagrammatically the arrangement of the oiling system at the First National Bank Building, Chicago. The storage tank containing the supply of engine oil is under air pressure at all times except during the short periods when it is being filled with oil from the filter. The air pressure on the surface of the oil forces it to a manifold on the engine from which it is distributed to the various oil cups. The oil flows from the different

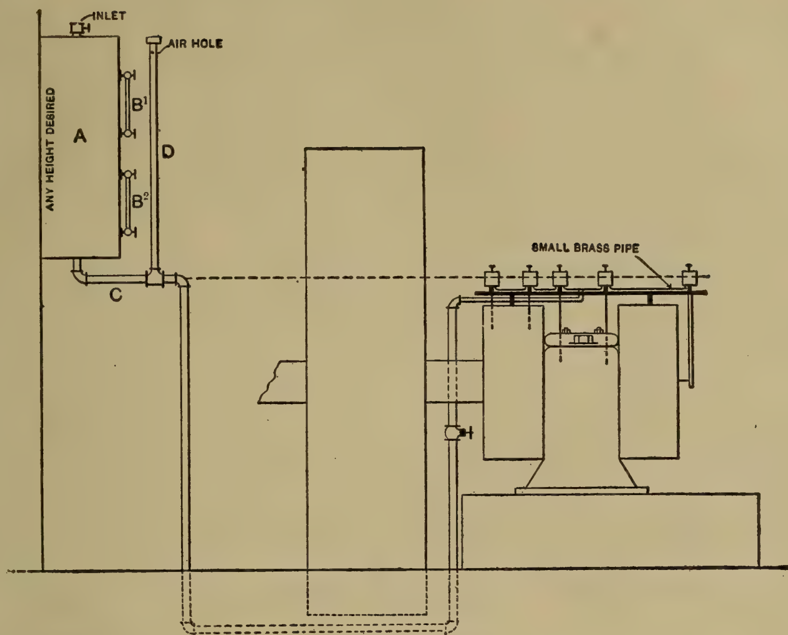


FIG. 530. Low-Pressure Gravity Feed, Constant Head.

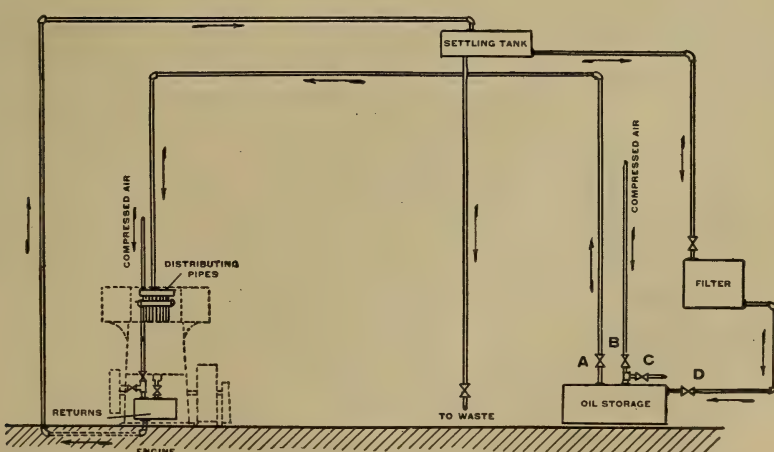


FIG. 531. Oiling System at the Power Plant of the First National Bank Building, Chicago

bearings to the returns tank located at the base of the engines. When the tank is filled air pressure is admitted and the oil forced to the settling tank, which has a capacity of about 400 gallons and is located near the ceiling. The oil is allowed to settle and the entrained water and foreign material are drained to waste. The oil gravitates from this tank to a series of Turner oil filters. When a new supply of oil is needed, valves *A* and *B* are closed and vent valve *C* opened, cutting off the supply of air and reducing the pressure to atmospheric. Valve *D* is then opened and oil flows from the filters to the storage tank.

422. Cylinder Lubrication. — The test requirements for cylinder oils are outlined in Table 126, from which it will be seen that pure mineral oil fulfills practically all requirements for dry steam. In connection with moist steam, as in the low-pressure cylinders of compound engines, an addition of from 2 to 5 per cent of acidless tallow oil is recommended. Vegetable oils, beeswax, lard oil, degras (wool grease), and the like should never be used in compounding cylinder oils. The best cylinder oils are made from Pennsylvania stock. For data pertaining to the amount and grade of cylinder oil used in a large number of piston engine plants see Table I, p. 824, Jour. A.S.M.E., May, 1910. See also "Lubricants and Lubrication," by Dr. C. F. Mabery, Jour. A.S.M.E., Feb., 1910.

Cylinder oils must be forced to the parts requiring lubrication against the prevailing steam pressure, which is ordinarily accomplished by (1) *cylinder cups,* (2) *hydrostatic lubricators,* or (3) hand or power driven *force pumps.*

423. Cylinder Cups. — A cylinder oil cup consists essentially of a steam-tight brass vessel fitted at the bottom with a pipe connection and valve. A screwed cap offers a means of introducing the lubricant into the cup. After the cap is in place the valve is opened and the cup is subjected to full steam pressure. The pressure in the cup, being equal to that in the steam chest or cylinder, permits the lubricant to gravitate through the valve into the cylinder.

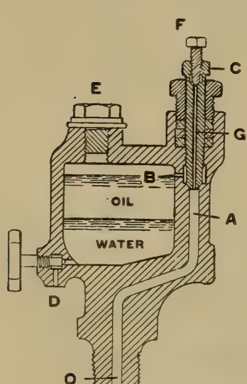


Fig. 532. Leyland Automatic Cylinder Cup.

Fig. 532 shows a section through an improved form of oil cup in which the oil feeds from the top instead of the bottom as is the case with the common form of cylinder cup. The vessel is attached to the steam chest or to the supply pipe below the throttle valve. Steam is admitted through opening *B* and, condensing, settles through the oil to the bottom. This raises the level of the oil until it

begins to overflow down the same passage by which the steam enters. This action is intensified by the fluctuation in steam pressure. The rate of feeding is regulated by valve *C* and tested by unscrewing plug *F*. If oil appears through opening *G*, the cup is feeding oil; if steam or water is emitted the cup is empty. The cup is filled by means of plug *E* and the water drained at *D*.

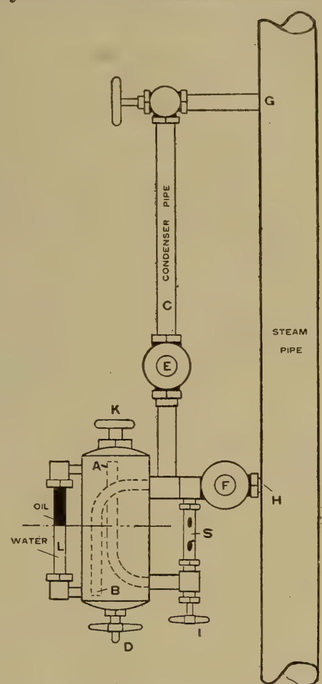


FIG. 533. Common Hydrostatic Lubricator.

424. Hydrostatic Lubricators. — The most common method of cylinder lubrication is by means of *hydrostatic* lubricators of the sight-feed class, Fig. 533. The principle of operation is as follows: The lubricator is filled with cylinder oil by removing cap *K*, the height of oil appearing in glass *L*. If water is present the oil floats on top as indicated. After the cap is screwed in place the valves in the condenser pipe are opened, subjecting the oil in the vessel to steam-pipe pressure. Steam is condensed in pipe *C*, filling tube *B* and part of *C*, thus adding to the steam pressure the pressure due to the weight of the water column. Valve *F*, which communicates with the top of the vessel by means of tube *A*, is opened wide, as is also the regulating valve *I*. The pressure at *B* being greater than that at *A* by an amount equivalent to the height of the water column, forces the oil through *A* and the "sight feed" *S* to the steam pipe. The rate of flow is controlled by the regulating valve *I*. As the oil flows from the vessel its space is occupied by condensed steam, the height of oil and water being visible in glass *L*. Owing to the small capacity of the lubricator it must be refilled frequently. To reduce the amount of labor required with the above apparatus, independent sight feeds, Fig. 409, are sometimes used in connection with a central reservoir. Such an installation is

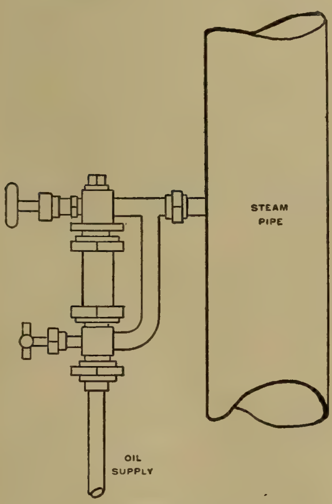


FIG. 534. Lunkenheimer Sight-feed Lubricator.

shown diagrammatically in Fig. 535. A condenser pipe leading from the steam main enters the bottom of the reservoir and the condensed steam fills up the reservoir as fast as the oil is fed out. The principle is the same as that of the simple hydrostatic lubricator. Oil is frequently injected by mechanical means under a steady pressure generated and governed independently of the steam. Two systems are in common use, direct mechanical pump pressure and air pressure.

425. Forced-feed Cylinder Lubrication. — Fig. 536 illustrates the "Rochester" simple feed automatic lubricating pump, which takes the oil by gravity from the reservoir through a sight-feed glass and forces it through a small pipe to the steam supply pipe. The pump entirely obviates the trouble due to intermittent feeding and, being directly

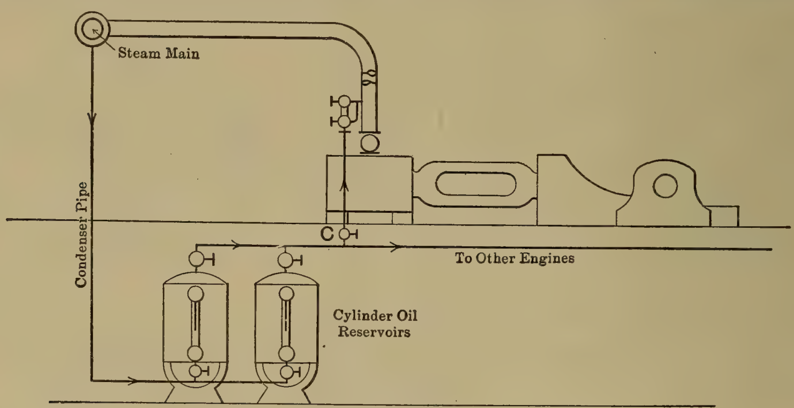


FIG. 535. Central Hydrostatic Lubricator.

driven from the engine, runs at constant speed. The feed is uniform and independent of the pressure pumped against. The rate is determined by the length of stroke of the pump piston, which is easily adjusted.

With large engines multi-feed pumps are sometimes used, which force oil to the various valves as well as to the steam pipe. Fig. 537 shows an arrangement of storage tank in connection with pump reservoir to avoid the trouble of hand filling.

426. Central Systems. — Fig. 538 shows the piping for a large central system of cylinder and engine lubrication. There are two storage tanks on the engine-room floor, one for cylinder oil and the other for engine oil, the distributing arrangements being the same in each case. The oil is pumped from each tank into a main pipe extending the length of the engine room and provided with branches at each point requiring lubrication. The oil pumps are actuated by steam and are of the duplex direct-acting type, provided with automatic governors which regulate

the speed to suit the demand for oil. The cylinder oil is forced through a special sight-feed lubricator, Fig. 539, under a pressure of about 25 pounds in excess of the steam pressure. Referring to Fig. 539, diaphragm valve *D*, in the bottom of the lubricator, is kept closed by

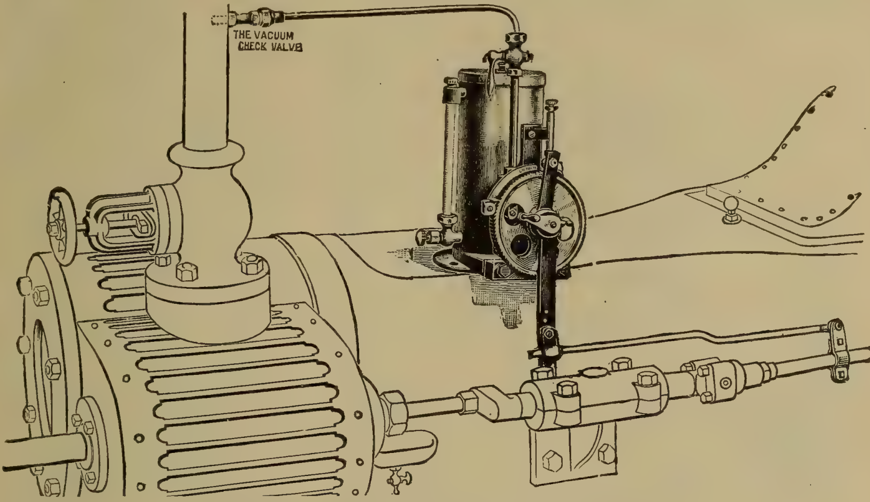


FIG. 536. Rochester Forced-feed Lubricator.

the steam pressure admitted through pipes *B*. Thus the inlet pressure must be greater than that of the steam before the valve will open and admit oil to the engine. The oil, after entering, passes upward through the sight-feed glass and downward through the hollow arm *A* to the

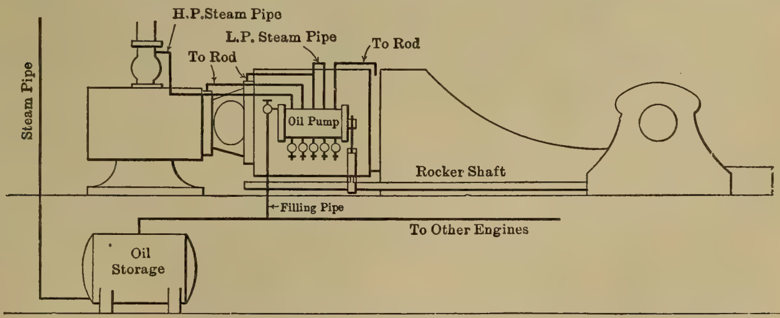


FIG. 537. Forced-feed Cylinder Lubrication.

steam pipe. The engine oil is forced by the pump to the various points under a pressure of about 20 pounds. The waste oil is caught in suitable receptacles and, after being filtered, is returned to the storage tank by a steam pump. This pump is connected so that it can supply the storage tank either from the filter or with fresh oil from a large oil tank

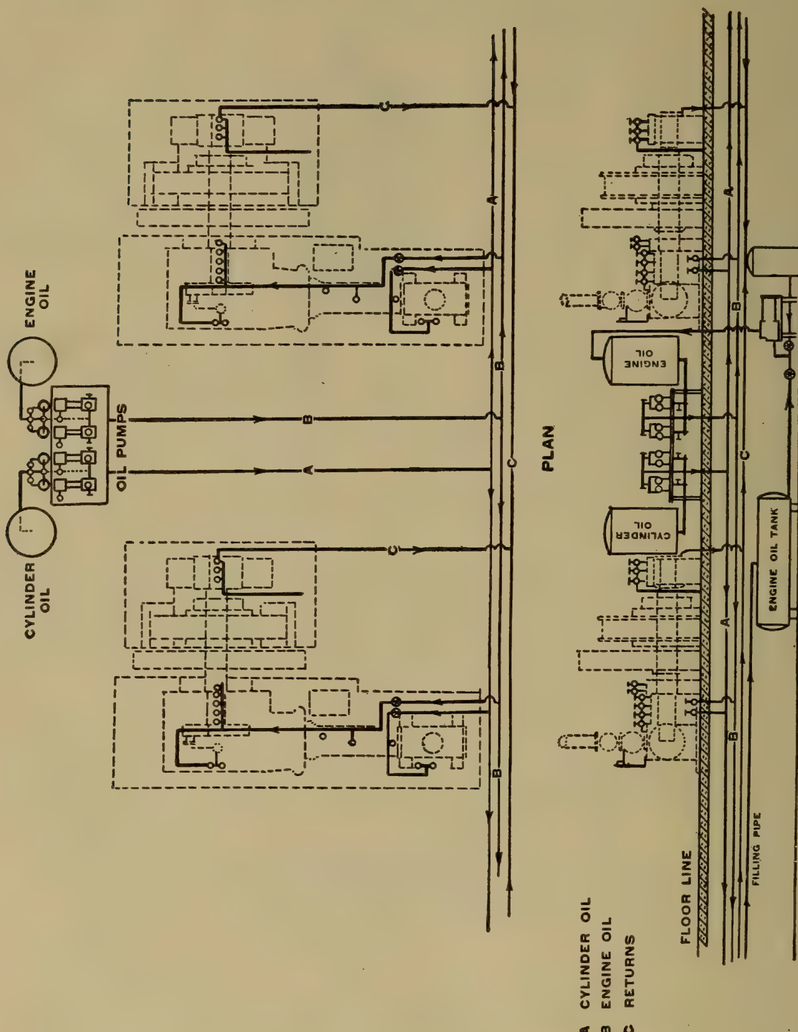


FIG. 538. Central System for Large Stations.

in the basement. By this arrangement all handling of oil in the engine room is done away with.

427. Oil Filters. — After oil has been applied to machinery its lubricating properties become impaired on account of (1) contamination with anti-lubricating material, such as dust, metallic particles from wear, gum, acid, and resin; and (2) exposure to heat and the atmosphere which drives off part of the more volatile constituents and decreases the fluidity of the oil.

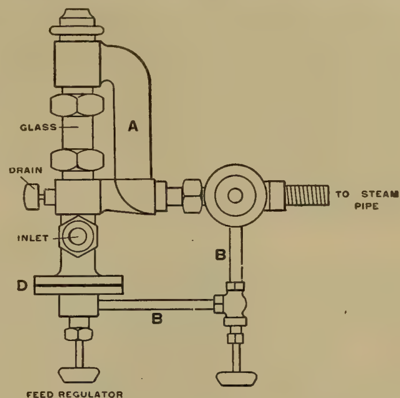


FIG. 539. Siegrist Sight-feed Lubricator.

In many small plants no attempt is made to reclaim oil that has once been used, since the quantity is so small that the cost and trouble involved would more than offset the gain. Where large quantities of oil are used, considerable saving may be effected by using it over and over again. To render the oil fit for reuse it must be thoroughly purified. The anti-lubricating matter is removed by precipitation and filtration.

Fig. 540 shows a section through a "White Star" oil filter and purifier. The apparatus consists of a cylindrical sheet-iron vessel divided into two compartments by a vertical partition. These two compartments are connected near the top by valve *B*. The smaller chamber is provided with a funnel *A* and a steam coil for heating the contents. The large chamber contains a cylindrical wire screen covered with several folds of filtering cloth. Impure oil is poured into funnel *A*, the upper part of which is provided with a removable sieve or strainer, and is discharged below the surface of the water through holes in the foot of the tube. The thin streams of oil rise vertically to the surface of the water and the heavy particles of grit and dirt gravitate to the bottom. The steam coil heats the oil and water and facilitates precipitation of

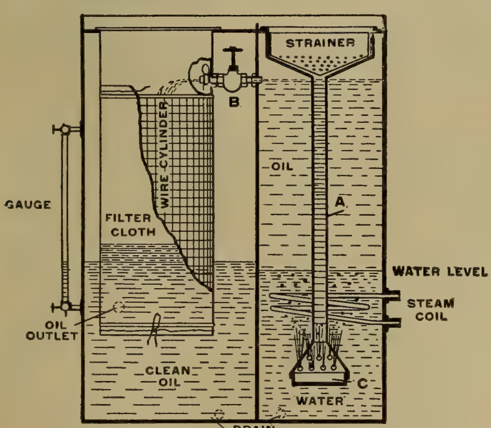


FIG. 540. White Star Oil Filter.

the solid matter by thinning out the streams of oil. When the oil in the smaller chamber reaches the level of valve B it flows into the filter bag, which removes the remaining impurities and permits the purified products to flow into the large compartment from which it may be drawn at will. All parts are accessible and readily removed for cleaning purposes. The accumulated sediment in the bottom of the small chamber is discharged to waste at intervals by means of a suitable drain. When the filter cloth is to be removed, valve B is closed and the wire cylinder is disconnected and lifted out. Any oil remaining in the filter is returned to funnel A. The filter cloth is held against the screen by cords and hence is readily removed.

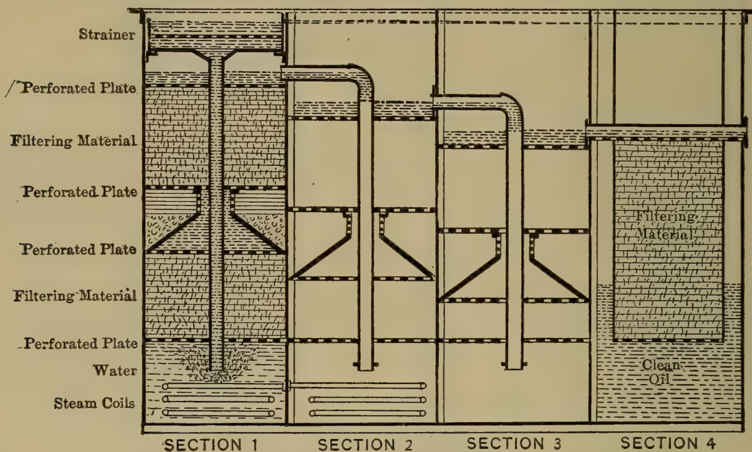


Fig. 541. Turner Oil Filter.

Fig. 541 shows a section through a Turner oil filter, illustrating the type of filter usually installed in large stations where continuous filtration is desired. This apparatus consists of a rectangular tank divided into four compartments. The returns from the lubricating system flow into section 1 through a screened funnel and discharge into the water space at the bottom of the compartment. The oil rises through the water, passes, under pressure of the head in the funnel, through a layer of filtering material resting on a perforated plate, and collects in an inverted cone. Through perforations around the top of the cone it passes into a dirt chamber, where most of the heavy impurities are deposited, and then, still rising, passes through another perforated plate and more filtering material. The partially cleaned oil, which issues, overflows into the second compartment and thence into the third, the same cycle of operations being repeated in these two. The overflow from the third compartment descends through a final

filter in the fourth compartment and collects at the bottom, from which it is withdrawn by the oil pump.

Cylinder Lubrication: Power, Feb. 15, 1910, Jan., 1905, p. 36; Engr., Lond., 1903, Vol. 96, pp. 55, 108, 132, 155; Engr. U. S., Oct. 15, 1906, p. 682; Jour. A.S.M.E., Feb. and May, 1910.

Miscellaneous. — Measurement of Durability of Lubricants: Trans. A.S.M.E., 11–1013. *Valuation of Lubricant by Consumer:* Trans. A.S.M.E., 6–437. *Suitability of Lubricants:* Power, Nov., 1906, p. 673. *Oil Required for Lubricators:* Elec. World, May 5, 1906, p. 934. *Gumming Tests:* Jour. Am. Chem. Soc., April, 1902, p. 467. *Valuation of Lubricants:* Jour. Soc. Chem. Ind., April 15, 1905, p. 315.

Lubrication, General: Power, Sept. 12, 1911, p. 396; Prac. Engr., Feb. 15, 1912, p. 194.

Oil Purification: Elec. World, Dec. 1, 1906, p. 1053.

Economy in Lubrication of Machinery: Trans. A.S.M.E., 4–315. *Theory of Finance of Lubrication:* Trans. A.S.M.E., 6–437.

Experiments, Formulas, and Constants for Lubrication of Bearings: Am. Mach., 1903, pp. 1281, 1316, 1350.

Lubricators and Lubricants: Power, Sept. 21, 1909, p. 486; Feb. 22, 1910, p. 347.

Selection of an Oil for Lubrication: Power, July 27, 1909, p. 137.

CHAPTER XVII.

TESTING AND MEASURING APPARATUS.

428. General. — The importance of maintaining a system of records is discussed in paragraph 459. The various items which may be recorded and the instruments and appliances used in this connection are outlined in the accompanying chart. In large stations a full complement of indicating, recording, and integrating instruments may prove to be a good investment if intelligently and closely studied by the operating engineer with a view to locating and eliminating unnecessary losses. The instruments should be inspected and calibrated at intervals, since many of them are delicately constructed and are apt to become inaccurate after a few months' service. Steam gauges, thermometers, and pyrometers, and particularly water meters are subject to appreciable error after considerable use. Voltmeters, ammeters, and other switchboard instruments are easily deranged, especially when subjected to continuous vibration or to high temperature.

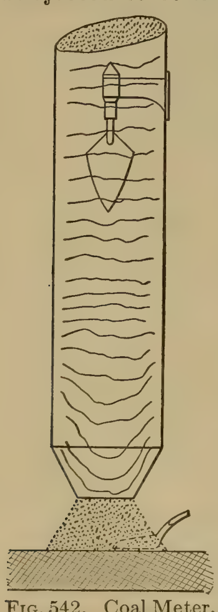


Fig. 542. Coal Meter.

429. Weighing the Fuel. — In most small plants the delivery tickets of the coal dealer are depended upon for the weight of coal used, no attempt being made to determine its evaporative value, and the economy of the plant is judged by the size of the coal bill. In such cases a considerable saving may be effected by keeping a daily record covering at least the coal and water consumption. The coal can be conveniently weighed on ordinary platform scales. In a number of large stations the weight of coal is determined by suspended weighing hoppers, which may be stationary, as in Fig. 148, or mounted on a traveling truck, as in Fig. 149. The scales of such devices are made indicating, autographic, integrating, or a combination of the three, the latter costing but little more than the simple indicating or recording devices.

A simple and inexpensive coal meter recently brought out is illustrated in Fig. 542. It consists essentially of a helical vane placed in a cylindrical conduit. The movement of the coal causes the vane to rotate and the number of revolutions is a measure of the weight of fuel pass-

TESTING AND MEASURING APPARATUS.

STEAM PLANT.

- Weights
  - Fuel
    - Platform scales, indicating and autographic.
    - Suspension hoppers, indicating and autographic.
    - Coal meters, integrating.
  - Water
    - Platform scales and tanks.
    - Water meters
      - Piston, Rotary, Disk — Indicating.
      - Venturi, indicating and autographic.
    - Weirs and volume displacement meters.
  - Steam
    - Weighing condensed steam.
    - Steam meters
      - Direct.
      - Indirect.
- Pressures
  - High
    - Bourdon gauge, indicating and autographic.
    - Manometers, mercurial, indicating.
  - Low
    - Manometers — mercurial, indicating, and autographic.
    - Manometers — water, indicating, and autographic.
    - Diaphragms, indicating and autographic.
- Temperatures
  - Up to 800 deg. F.
    - Mercurial thermometers, indicating.
    - Expansion thermometers, indicating and autographic.
  - 800 to 2500 deg. F.
    - Expansion thermometers, indicating and autographic.
    - Resistance thermometers, indicating and autographic.
    - Thermo-electric thermometers, indicating and autographic.
  - Over 2500 deg. F.
    - Optical pyrometer, indicating and autographic.
    - Platinum or clay ball pyrometer.
- Power
  - Indicated
    - Indicators, hand manipulated.
    - Indicators, continuous autographic.
  - Developed
    - Rope brake.
    - Prony brake.
    - Absorption dynamometers.
    - Electric generator.
- Flue gas analysis
  - Orsat apparatus.
  - Arndt's econometer, indicating.
  - Westover recorder, autographic.
  - Uehling gas composimeter, autographic.
- Moisture
  - In air . . . Hygrometer, indicating and autographic.
  - In steam . . . Calorimeters
    - Separating.
    - Throttling.
- Fuel analysis
  - Coal calorimeters
    - Mahler bomb.
    - Carpenter.
    - Thompson.
    - Parr.
  - Gas calorimeter . . . Junker.

ELECTRICAL PLANT.

Voltage	Voltmeters, A. C. and D. C., indicating and autographic.
Current	Ammeters, A. C. and D. C., indicating and autographic.
Output	Wattmeters, A. C. and D. C., integrating and autographic.
Power factor	Power factor meters, A. C. only, indicating and autographic.
Frequency	Frequency meter, A. C. only, indicating.
Synchronism	Synchronizers, A. C. only, indicating.

ing. For hard coal of uniform size the meter gives consistent results agreeing within two per cent of scale weight, but with bituminous coal the results are somewhat erratic and particularly so with lumps of varying size. (For a detailed description of the device, see Prac. Engr., U. S., Apr. 15, 1912, p. 438.)

430. Measurement of Feed Water. — The quantity of water fed to the boiler may be determined by

1. Actual weighing.
2. Measurement of volume displacement.
3. Measurements by weirs and orifices.
4. Measurement by determining the velocity of flow in the feed pipe.

Some of these methods necessitate measurement on the suction side of the pump; others are applicable to either suction or pressure. The former, as a class, are the more accurate but involve bulky apparatus. The choice for any given case depends upon the quantity of liquid to be measured, the degree of accuracy required, space requirements, and first cost.

431. Actual Weighing of Feed Water. — The most accurate means of measurement is by the use of two or more tanks resting upon scales, arranged to be filled and emptied alternately. This method is limited to comparatively small quantities because of the great bulk of apparatus involved and is seldom used for continuous service. It is commonly employed in conducting special tests of short duration and for calibration purposes. For regular boiler service it involves considerably more time than is ordinarily at the disposal of the fireman and engineer. For temperatures above 150 degrees F., the weighing tanks should be covered, since evaporation may cause an appreciable error. See also "Rules for Conducting Boiler Trials," A.S.M.E., Code of 1912, reprinted in Appendix B.

432. Worthington Weight Determinator. — Fig. 543 shows the general details of the Worthington weight determinator, illustrating a commercial means of continuously measuring and recording the *weight* of water fed to the boiler. The apparatus consists primarily of two tanks of equal size, A and B, each mounted on knife edges K and equipped at one end with a siphon S and at the other end with counter weight W. The liquid to be measured flows through inlet pipe P and along deflector D into either tank. Each tank remains in a horizontal position until the weight of liquid overcomes the counter weight when it tilts into the position shown by the dotted lines. Discharge now takes place through siphon S until the liquid reaches a certain level at which point the tank tilts back to its original position and the siphon continues its action until the vessel is emptied. The tanks operate

alternately, one filling while the other is discharging. Since each tilt represents a definite *weight* of liquid irrespective of variations in volume due to specific gravity or changes in temperature, the number of tilts as recorded by counter C is a correct measure of the weight discharged. This apparatus operates at atmospheric pressure and is arranged to discharge into a storage tank from which the feed pump takes its supply.

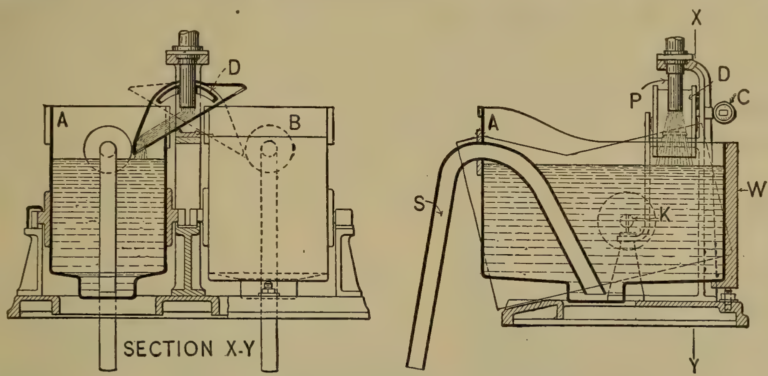


Fig. 543. Worthington Water Weigher.

433. Kennicott Water Weigher. — This apparatus is used in many boiler houses and seems to give universal satisfaction. It consists of a cylindrical shell S, Fig. 544, the lower part of which is divided into two measuring compartments A and B, each fitted with a siphon for discharge and a float F for actuating the tripping mechanism. Tripping box E is divided into two sections which alternately fill with water and serves the double purpose of furnishing a sufficient quantity of water to start the siphons and to shift the supply from one compartment to the other. This tripping box is balanced on knife edges and is mounted directly above the measuring compartments. Water enters the inlet and passes to the tripping box where a small portion is intercepted, the remainder passing directly to the measuring compartment below. When this compartment is nearly filled the float tilts the tripping box, discharges its contents into the compartment, and starts the siphon. A counter registers each double charge. This apparatus discharges at atmospheric pressure, though with slight modification it may be installed on the pressure side of the pump. Kennicott water weighers are constructed in various sizes ranging from a capacity of 750 to one million pounds per hour and are guaranteed by the manufacturers to record the correct weight of water within one-half of one per cent of scale weight at any given temperature. Calibration for different temperatures is necessary since the apparatus is actuated by volume displacement. For example, the weight of one cubic foot of water at

60 degrees F. is 62.37 pounds and at 210 degrees F. it is 59.88, a difference of 2.49 pounds. Hence, if the device is calibrated to read correctly at 60 degrees it would be in error 4 per cent if used to measure water at 210 degrees F.

434. Willcox Water Weigher. — Another successful volume displacement meter is illustrated in Fig. 545. The device consists of a cylindrical tank divided into an upper and lower compartment by a horizontal partition. The water enters the upper compartment, passes to the lower, in which its volume is measured, and then out through the U-shaped discharge pipe. The operation, beginning with both compartments empty, is as follows: Water enters the upper compartment

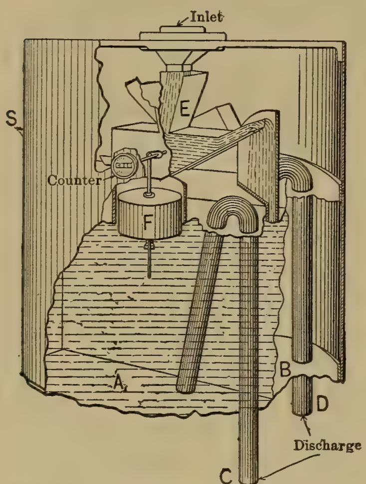


Fig. 544. Kennicott Water Weigher.

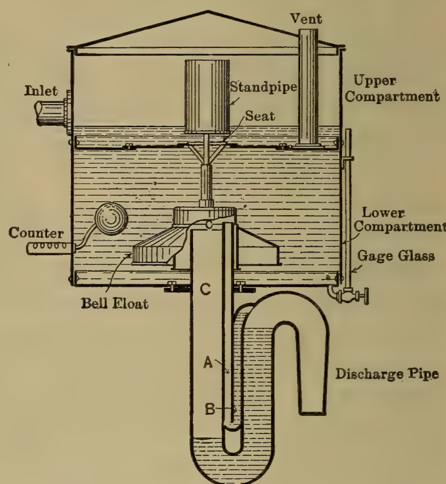


Fig. 545. Willcox Water Weigher.

through the inlet pipe and rises to the top of the standpipe. (The latter is open at the top and bottom and is rigidly connected to the bell float, but when in its lowest position it is held against its seat by weight of the bell float.) Further admission of water causes it to overflow into and through the standpipe into the lower compartment. The water, rising in the lower compartment, seals the lower edge of the bell float and entraps a volume of air under the bell. Further rise compresses the air under the float, in leg C of the discharge pipe and in leg A of the trip pipe AB. This compression causes the float to rise to its highest position and raises the standpipe from its seat, permitting the water in the upper chamber to pour into the lower vessel. Compression of air continues until the pressure becomes great enough to break the seal in the trip pipe. This action immediately reduces the pressure below the float, permits the latter to descend, sealing the upper chamber against

further discharge, and allows the water in the lower compartment to siphon out through the discharge pipe. The number of discharges is recorded mechanically.

435. Weir Measuring Devices. — Feed water heaters or specially designed tanks fitted with V-shaped or trapezoidal weir notches offer a simple means of measuring the instantaneous rate of flow. The heater chamber is divided into vertical compartments arranged so that one may discharge through a calibrated weir notch into the other. The height of water above the bottom of the notch is a direct measure of the volume flowing. The height may be noted in an ordinary gauge glass or it may be transferred through a suitable float mechanism to an outside indicator. In determining the total flow for any given period

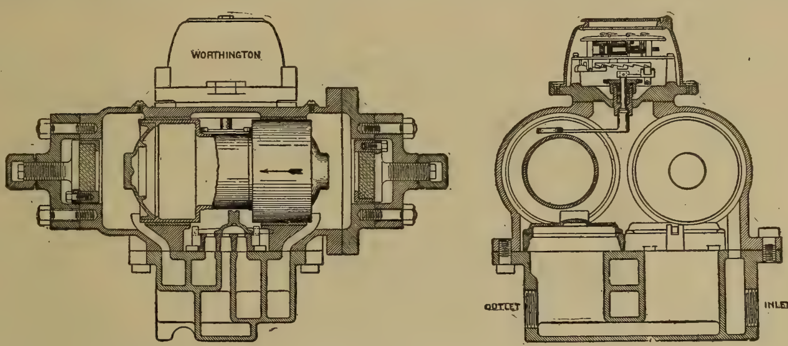


Fig. 546. A Typical Piston Water Meter. (Worthington.)

it is necessary to take readings at frequent intervals. The chief drawback to this method lies in the difficulty of preventing sudden fluctuations in water level due to surging in the heater. For the theory of weir notches, orifices, and nozzles consult "Experimental Engineering," Carpenter and Diederichs, 1911, Chapter XII. See also, Jour. A.S.M.E., Oct., 1912, p. 1479.

436. Pressure Water Meters. — There are a number of reliable water meters on the market for hot or cold water which may be placed on the pressure side of the feed pump. Among them may be mentioned the Hersey, Crown, Nash, and Worthington. They are all based on volume displacement and consequently require correction for different temperatures if graduated to read in pounds. They are compact, comparatively inexpensive, and require considerably less space than the tank weighers of the Kennicott and Willcox type but are open to the objection that no particular provision is made against leakage and after considerable use they are subject to serious error. In many plants where meters of this type are installed the meter is by-passed and operated only for short periods. For continuous service meters of the tank-

weighing or Venturi type are recommended. Fig. 546 illustrates the *piston* type of pressure meter, in which reciprocating pistons are displaced by a definite volume of water; Fig. 420, the *rotary* type, depending upon the displacement of rotary impellers; Fig. 547, the *disk* type, in which impellers are given a combined rotating and tilting

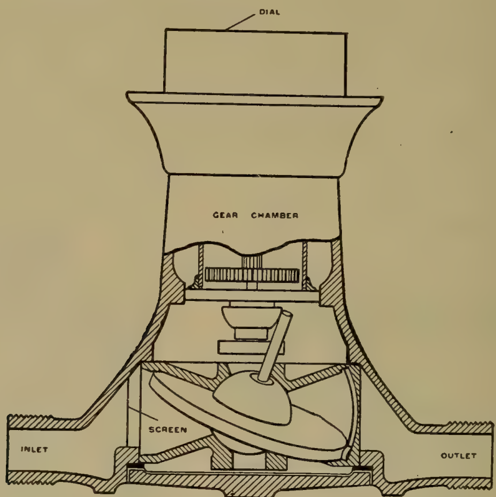


FIG. 547. A Typical Disk Water Meter. (Nash.)

motion. The capacities of pressure meters, irrespective of type or maker, range approximately as follows:

Size of meter (pipe size)	$\frac{3}{8}$,	$\frac{1}{2}$,	$\frac{3}{4}$,	1,	$1\frac{1}{2}$,	2,	3,	4,	6
Maximum capacity, cubic feet per minute:									
Rotary or disk meters	1,	2,	4,	8,	12,	20,	36,	72,	120
Piston meters		$1\frac{1}{2}$,	3,	5,	6,	8,	23,	60,	120.

437. Venturi Meter. — The Venturi tube with indicating, autographic, and integrating mechanism, as constructed by the Builder's Iron Foundry of Providence, R. I., is one of the most satisfactory methods of measuring feed water under pressure. The total absence of working parts in the meter proper insures continuity of operation and freedom from wear, and the fact that the recording mechanism may be placed at a considerable distance from the meter is a great advantage. The Venturi tube, Fig. 548, is essentially the same in principle as an orifice placed in the pipe. The pressure difference H between A in the "upstream" portion of the tube and B at the "throat" is a measure of the velocity through the throat. The loss of head due to friction is negligible and the velocity may be calculated, within an

error of 3 per cent, from the following modification of Bernouilli's theorem:

$$V_t = \frac{F_u}{\sqrt{F_u^2 - F_t^2}} \sqrt{2\,gH}, \tag{284}$$

in which

V_t = velocity at the throat, feet per second.
F_u = area of the upstream section, square feet.
F_t = area of the throat, square feet.
H = pressure difference, feet of water.

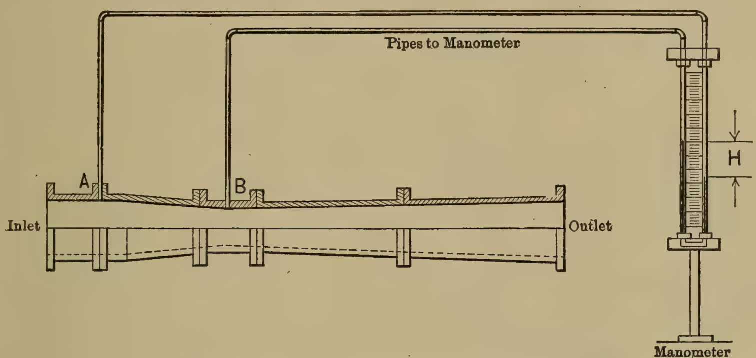


FIG. 548. Venturi Tube with Indicating Manometer.

For accurate work the tube requires calibration. Once calibrated the error in weight readings for a given temperature should not exceed one per cent for capacities within the working range of the manometer. For very low throat velocities the error may be considerable because of the slight pressure difference between the points A and B. In situations where there are periods of very low and very high rates of flow, as in connection with combined heating and lighting plants, it is customary to install a small tube for the light loads and a large tube for the heavy loads, the same indicating mechanism being used in each case. The equipment illustrated in Fig. 548 is purely indicating and readings must be taken at frequent intervals in order to obtain the total flow for a given period. Where the size of the plant warrants the outlay the combined indicating, integrating, and recording instrument is often installed. With this device the instantaneous rate of flow is indicated by a pointer and dial, the variation in rate of flow for any given period is recorded on a clock-driven chart, and the total weight flowing is registered on a counter. (For a detailed description of this mechanism see Power, Jan. 23, 1912, p. 102.) Tests made at Armour Institute of Technology on a carefully calibrated tube and recorder with feed water at 210 degrees F. and constant rate of flow gave chart

and counter readings agreeing substantially with scale weights; for irregular and fluctuating flow, as when feeding the boilers, the average error was about one per cent.

438. Orifice Measurements. — The appropriation of the great majority of small steam power plants does not permit of the installation of tank meters, Venturi meters, or other forms of reliable commercial appliances for measuring the weight of water fed to the boilers. For use in such cases an inexpensive and fairly accurate indicating meter may be constructed of ordinary pipe fittings, as illustrated in Fig. 549. A thin metal diaphragm with circular orifice is inserted on the pressure side of the feed pump and the pressure drop across the orifice is measured by inclined mercury manometer. The height of mercury

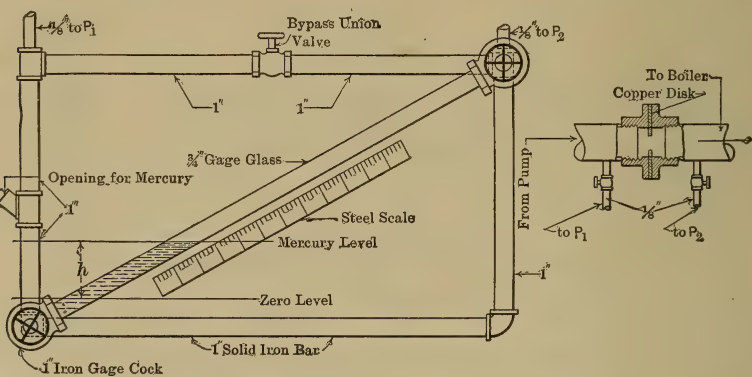


FIG. 549. Simple Indicating Water Meter, Orifice Type.

h is an indication of the rate of flow. By calibrating the manometer against tank measurements the readings of the mercury column may be graduated to read directly in pounds per hour. If means are not available for calibration purposes the weight of discharge may be approximated from the formula

$$W = 1120\, a \sqrt{hd}, \tag{285}$$

in which

W = weight flowing, pounds per hour.
a = area of the orifice, square inches.
h = vertical height of mercury column, inches.
d = density of the water, pounds per cubic foot.

For a fairly continuous flow and pressure drop corresponding to three inches of mercury or more this simple device gives results agreeing within four per cent of tank weights, but for widely fluctuating flow and small pressure drops the error may be considerably more.

For application of the Pitot tube for water measurements consult accompanying bibliography.

The Pitot Tube for Water Measurements: Trans. A.S.M.E., Vol. 30, 1908, p. 351; Vol. 25, 1904, p. 184; Vol. 22, 1901, p. 284.

The Pitometer: Proc. Am. Water Wks. Asso., 1907, p. 136; Jour. Frank. Inst., Dec., 1907, p. 425.

439. Measurement of Steam. — The quantity of steam passing through any device may be determined by (1) condensing and weighing the steam after it has passed through the apparatus and by (2) measuring the flow by means of steam meters before it enters. The first necessitates the use of surface condensers, and consequently has a limited field of application, whereas the latter may be used in both condensing and non-condensing service.

440. Weighing Condensed Steam. — The weight of condensed steam may be obtained by any of the devices used in connection with feed water measurements but such measurements are seldom made except for test purposes because of the expense or labor involved. The Wheeler Condenser and Engineering Company's "indicating hot well" offers a practical and simple solution of continuously measuring the condensed steam. The hot well is attached to the bottom of the condenser chamber in the usual way and differs from the ordinary hot well only in the addition of a vertical partition. This partition divides the hot well chamber into two compartments. Condensation from the condenser drains into one of these compartments and flows to the other through a calibrated orifice. The height of water above the orifice as shown in the gauge glass is an indication of the weight of condensation flowing. The manufacturers guarantee an accuracy within 2 per cent of scale weight for readings over the whole range. The readings are purely indicating and must be taken at frequent intervals in order to give the total flow.

441. Steam Meters. — The weight of fluid flowing through an opening may be calculated by the equation

$$W = AyV, \tag{286}$$

in which

W = weight in pounds per second.

A = cross-sectional area in square feet.

y = density of the fluid, pounds per cubic foot.

V = velocity of flow, feet per second.

All steam meters for indicating or recording the weight of steam flowing through a pipe are based upon the law expressed in equation (286). Thus, for steam of constant density the opening through which it flows may be made constant and the variation in velocity will be an

indication of the rate of discharge; or the velocity may be held constant and a variation in the amount of opening will be an indication of the weight discharged. Unfortunately, the density of steam is seldom constant under commercial conditions and herein lies the inherent defect of all steam meters which depend for their operation upon a variation in the area of efflux or a variation in velocity. The density of steam is a function of its pressure and quality and any variation in either will affect the weight of discharge as determined from equation (286). Pressure variations may be automatically compensated for, but corrections for quality must be made in each specific case.

CLASSIFICATION OF STEAM METERS.

Indirect	Velocity	Pitot tube	Impeller	Lindenheim (1896)* Gebhardt (1908)†
			Water manometer	Burnham (1905)† Clyde (1910)†
			Mercury manometer	General Electric (1910)*†
		Current	Impeller	Holly (1877)*
Direct	Throttling	Floating valve	Mechanical control	St. Johns (1893)†‡ Gehre (1896)†‡ Baeyer (1902)†‡ Bendemen (1902)†‡ Sargent (1908)† Lindmark†‡
		Stationary disk	Mercury manometer	Gehre-Hallwachs (1907–1910)*†‡ Sarco (1910)*†‡ Storrer (1910)†‡
		Venturi tube	Bourdon manometer	Eckardts (1903)†‡
			Mercury manometer	Parenty (1886)†‡ Builders' Iron Foundry (1910)†‡

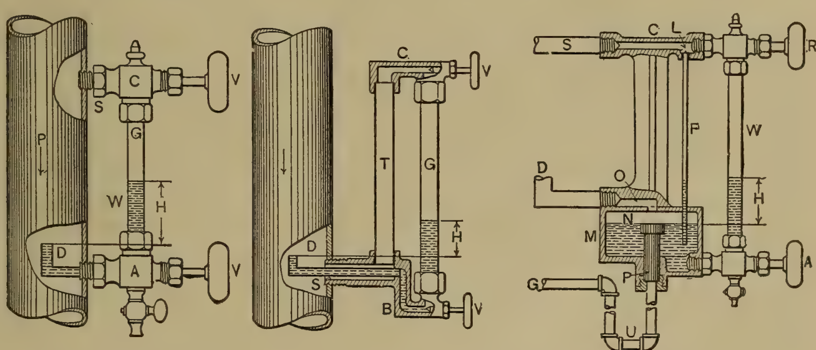
The different means adopted for transmitting this area and velocity variation to the indicating or recording devices overlap to such an extent as to render a classification of steam meters very unsatisfactory. The accompanying chart is offered as a guide in grouping the most commonly known devices. From this chart it will be seen that all meters may be grouped into general classes, *direct* and *indirect*. The direct meter is an integral part of the piping and the entire mass of

* Integrating. † Indicating. ‡ Autographic.

fluid to be measured passes through the apparatus. It is not portable and cannot be readily applied to pipes of different sizes. In the indirect meter only a small part of the fluid to be measured is directed through the apparatus and the pipe line need not be disconnected for its installation. One instrument suitably calibrated may answer for any size of pipe.

The average high-grade steam meter is a reliable and accurate means of measuring the flow of steam in straight lengths of pipes, provided the flow is continuous or that the change in the rate of flow is gradual and the pressure and quality are practically constant. For interrupted or intermittent flow and for sudden variations in pressure or quality, the results are not reliable and may be considerably in error. The accuracy of all meters, provided they have been correctly calibrated and adjusted, depends largely upon the degree of refinement in reading the indicators and in integrating the charts. The commercial failure of many steam meters is due to the fact that they are not cared for or operated in strict accordance with the principles of design.

Only a few of the best-known meters will be described here. For a detailed discussion of the various types of steam meters see the author's paper "Various Types of Steam Meters," Power, Feb. 6 and 13, 1912.



FIGS. 550, 551, 552. Principles of the "Gebhardt" Indicating Steam Meters.

442. "Gebhardt" Steam Meters. — Figs. 550 to 554 illustrate various forms of indicating steam meters designed and tested at the Armour Institute of Technology, which are based on the principles of the Pitot tube. Referring to Fig. 550, A and C are two ordinary gauge cocks and G is a common gauge glass, C being connected with the static nozzle S and A with the dynamic tube D. The height of water H is proportional to the square of the velocity of steam flowing through pipe P and automatically adjusts itself to the variations in velocity; thus, for decreasing velocities, the water in glass G discharges through

D until the water column H balances the velocity pressure in pipe P, and for increasing velocities, condensation from the upper part of the instrument accumulates and the water column H rises until a balance is effected for the higher velocities.

The relation between the height of the water column and the velocity of the steam in the main pipe at the entrance to the dynamic tube may be determined from the well-known equation

$$V = c\sqrt{2\,gh}, \tag{287}$$

in which

V = maximum velocity of flow, feet per second.
c = coefficient determined by experiment.
h = height of a column of steam equal in weight to the water column H.

This equation may be expressed

$$V = K\sqrt{H\frac{d_w}{d_s}}, \tag{288}$$

in which

K = coefficient determined by experiment.
H = height of water column in inches.
d_w = density of water in gauge glass, pounds per cubic foot.
d_s = density of steam in the main pipe.

Because of the labor of determining the relationship between the mean and the maximum velocity for various conditions of flow and different pipe diameters it is more satisfactory to calibrate the gauge, by actual experiment, to read directly in pounds per hour.

This simple device in connection with a calibrated scale gives readings within 5 per cent of condenser measurements for continuous flow and constant pressure and quality of steam (for velocity pressures corresponding to $1\frac{1}{2}$ inch of water or more). For a considerable variation in pressure and quality or for marked changes in rate of flow the instrument is not reliable. Its sensitiveness is greater at high velocities, since the height of water column in the gauge glass increases with the square of the velocity of the steam in the main pipe. For interrupted flow, as when connected to a high-speed engine, the water column may be made to closely approximate the mean velocity by suitably throttling the gauge cocks.

Figs. 551 and 553 show application of the same principle with only one connection to the main pipe. The commercial meter (Fig. 553) gives readings within 2 per cent of condenser weights for velocity pressures corresponding to 1 inch of water or more. Fig. 552 shows another form which may be placed below or above the point in the main pipe at which the Pitot tubes are placed. The operation is as follows:

Velocity pressure is transmitted through tube D and opening O, into the body of the chamber M. This pressure, acting on the surface of the condensed steam in the chamber, forces the water into the glass W until a balance is effected. Condensation is discharged continuously through pipe P and the water seal U of the main pipe. Tests of this meter have given results agreeing within 2 per cent of condenser measurements for continuous flow for all velocities ranging from the equivalent of a 1-inch to a 10-inch water column. No provision is made for automatic correction of pressure and quality variation in any of these devices. (For the theory and results of tests of the Pitot type of steam meter see author's paper "The Pitot Tube as a Steam Meter," Trans. A.S.M.E., Vol. 31, p. 603.)

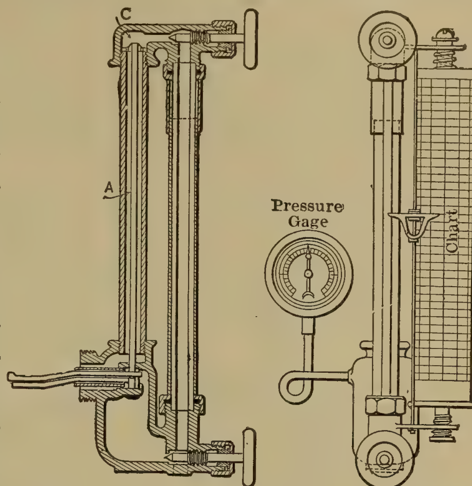


FIG. 553. Commercial Form of "Gebhardt" Steam Meter.

Fig. 554 shows a modified form for very low velocities or for low-density steam. The rise and fall of the water column is transmitted through a small float and multiplying gears to an indicating dial and pointer. Referring to the illustration, S is the static tube and D the dynamic tube. The movement of float B is transmitted through levers L and N to sector G and pinion P. Pinion P is secured to bar magnet M, so that the rise and fall of float B cause the magnet to rotate. Opposite magnet M, but outside the casing, is another bar magnet M' (see figure in lower corner of illustration). The movement of bar M within the casing is transmitted to bar M' outside the casing, magnetically, thus obviating the use of stuffing boxes.

443. General Electric Steam Meters. — The General Electric Company has recently placed on the market a number of steam meters of the Pitot-tube type in which a mercury column is used to measure the velocity pressure. The principle involved is an old one, but this company is the first to exploit it successfully from a commercial standpoint. Three styles are manufactured, (1) in which the velocity pressure is measured directly by means of a U-tube manometer; (2) in which the variation in the height of the mercury is transmitted to an indicating dial through the agency of floats and pulleys; and (3) in which the

variation in the weight of the mercury column actuates a recording mechanism by means of a series of compound levers.

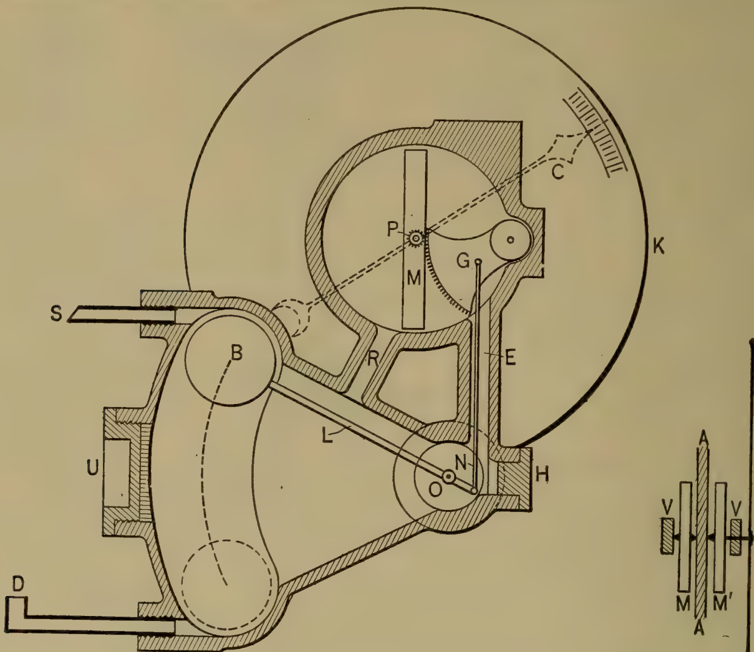


FIG. 554. "Gebhardt" Steam Meter for Low Velocities.

The principle of the simple indicating device is illustrated in Fig. 555, in which S is the static nozzle at right angles to and D the dynamic nozzle facing the current; U is an ordinary U-tube manometer partially filled with mercury. When there is no flow the surface of the mercury in the columns N and W will be on the same level and the upper portions will be filled with condensed vapor. When there is a flow, the mercury will be depressed as indicated and the difference H in the heights of the mercury columns will be a measure of the velocity of flow at the point in the pipe where the dynamic tube is placed.

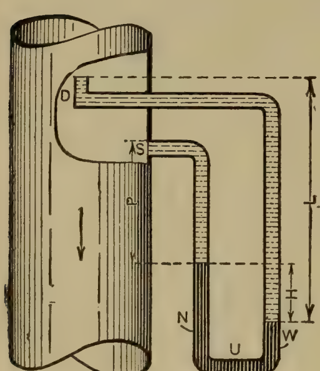


FIG. 555. Pitot Tube with Mercury Manometer.

This velocity may be expressed by substituting the proper values in equation (287), thus

$$V = K\sqrt{h\frac{d_m}{d_s}}, \qquad (289)$$

in which

d_m = density of the mercury in pounds per cubic foot. (Other notations as in equation (288)).

A comparison of equations (288) and (289) will show that the mercury manometer is less sensitive than the water manometer by an amount equivalent to $d_m \div d_w$ or approximately 13.6. The variable height of the water column above the mercury is usually included in the value of the coefficient K.

The General-Electric indicating-flow meter, shown in Fig. 556, differs from the simple device just described in that a simple nozzle plug, shown in detail in Fig. 557, is used in place of the ordinary static and dynamic nozzles. Referring to Fig. 557, TT are the static openings or "trailing set" and LL the dynamic openings or "leading set." The plug is screwed into the pipe with the "leading set" directly facing the current and the connections to the manometer are made through the openings T' and L'. The weight of steam flowing may be obtained directly from the height of the mercury column H, Fig. 556, by means of suitable charts based upon experiments. Adjustment for variations in pressure, quality, and pipe diameter are made by setting the chart cylinder C in accordance with the graduated scales at the bottom of the instrument.

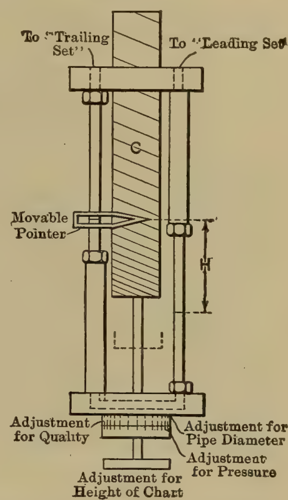


FIG. 556. General Principles of the G.-E. Indicating-flow Meter.

For general purposes a single revolving chart is furnished, the readings of which, multiplied by the area of the pipe, give the weight of steam

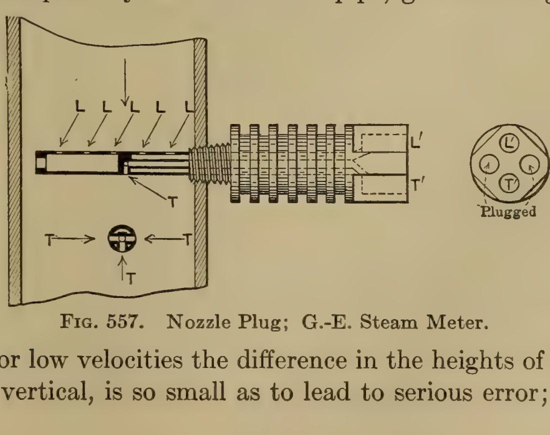


FIG. 557. Nozzle Plug; G.-E. Steam Meter.

flowing. For low velocities the difference in the heights of the mercury columns, if vertical, is so small as to lead to serious error; hence, pro-

vision is made for this by inclining the manometer as indicated in Fig. 558. With this the actual head of mercury due to the velocity is H, but the difference in the lengths of the columns is D. The indication on the chart corresponding to the height of the mercury in the glass T'' multiplied by a constant depending upon the inclination of the chart is the rate of flow in pounds per hour per square inch of the pipe cross-section.

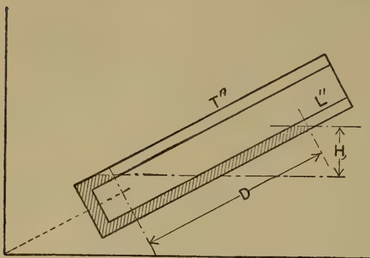


Fig. 558. Inclined Mercury Manometer.

The accuracy of this meter depends entirely upon the refinement of adjustment and the extent of error in reading the height of the mercury column. If correctly set, an error in reading of $\frac{1}{16}$ inch for a velocity corresponding to a vertical column $\frac{1}{2}$ inch high would be $12\frac{1}{2}$ per cent, whereas the same error referred to a vertical column $6\frac{1}{4}$ inches high would be but 1 per cent. Tests of this instrument, conducted at the Armour Institute of Technology, gave readings for continuous flow agreeing within 1 to 8 per cent of condenser measurements, depending upon the rate of flow. For interrupted flow the departure from condenser readings was more marked. This meter is portable, simple in construction, and readily applied to any pipe by inserting the nozzle plug at the required point, although considerable care is necessary in setting it up and in making the necessary adjustments for pipe diameter, steam pressure, and quality.

The G.-E. Boiler-flow meter differs from the one just described in that the variation in height of the mercury column is transmitted through a float and pulleys to two bar magnets, one within the casing and the other without. The indicating needle is fastened to the outer magnet and revolves in harmony with the variation in height of the mercury column. The magnet controlling device is identical in principle with that described in connection with Fig. 554.

The autographic instrument, Fig. 559, is one of the most successful recording devices on the market and is finding much favor with engineers. Its operation is shown diagrammatically in Fig. 560. Two cylindrical mercury cups R and R', constituting the legs of a U-tube mercury manometer, are pivoted on knife edge P and are connected to the nozzle plug by the flexible tubes A and A'. The instrument may be placed at any point below the level of the nozzle plug, one instrument being sufficient for a number of nozzles. When there is no flow the mercury in the two wells is at the same level and the system is in perfect balance. When there is a flow the velocity pressure causes the

mercury to flow from R to R' and the latter is lowered, carrying with it the recording pen C. The vertical distance on the chart between O and

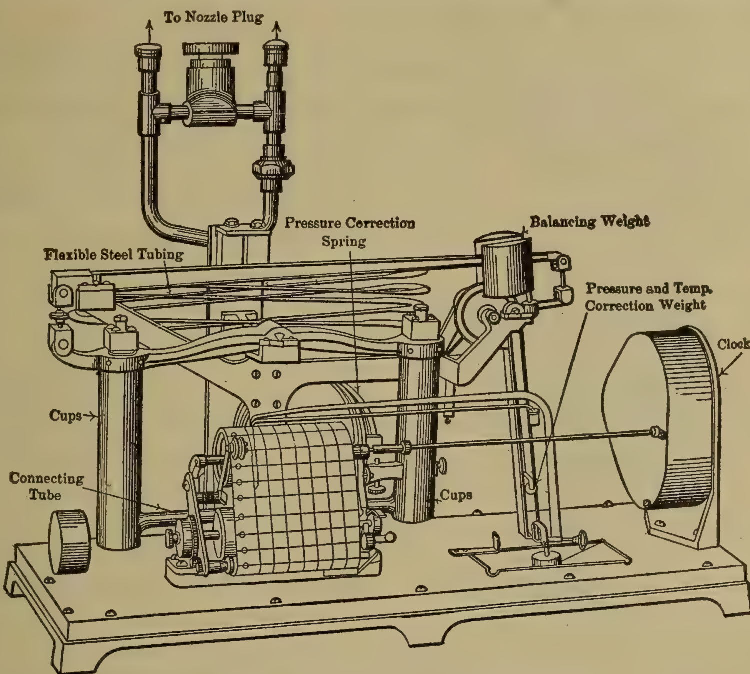


FIG. 559. G.-E. Recording Steam Meter.

C is a measure of the weight of steam flowing. (This system of mercury wells was used as early as 1886 in a steam meter designed by Parenty.)

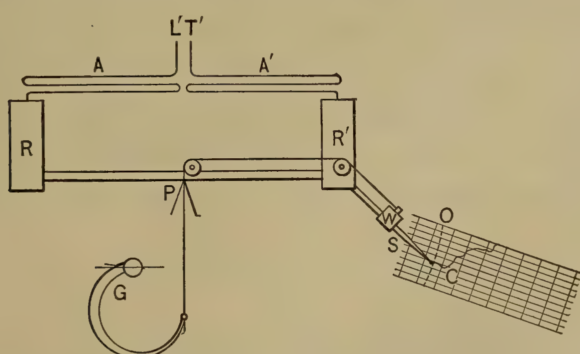


FIG. 560. Principles of the G.-E. Recording Steam Meter.

Pressure variation is automatically compensated for by the sliding weight W, the position of which relative to the knife edge is effected by the Bourdon tube G; thus for increasing pressures the gauge tube tends

to straighten out and raise the weight, thereby decreasing the leverage of well R'. Adjustment for quality is made by shifting the position of sliding weight W by hand.

In Europe the Gehre-Hallwachs meter, based upon the same principles as the G.-E. recorder, is fitted with an integrating device, thus enabling the total quantity of flow to be read directly. In the latter instrument the wells are cone shaped and proportioned so that the leverage increases directly with the weight of the steam flowing instead of increasing with the square of the weight as with the G.-E. cylindrical cups. This uniform motion of the cups makes it comparatively easy to add an integrating mechanism. Referring to Fig. 561, R is a small friction wheel mounted on the pen arm a and connected to gears c and d by the small shaft m; P is a clock-driven disk in contact with the friction wheel R. As the pen arm moves the wheel R in and out from the center of disk P, the speed of the small friction wheel is decreased or increased accordingly. The revolutions of R are transmitted to the integrating mechanism e so that the total flow may be read directly from the dials.

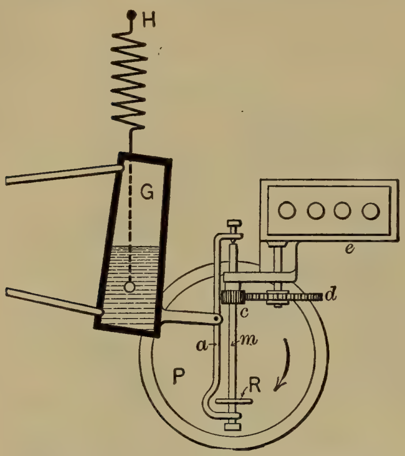


Fig. 561. Counting Mechanism for Steam Meters.

444. St. Johns Steam Meter. — In the groups of meters described above the indicating and recording mechanism is actuated by the natural velocity of the steam. In the St. Johns, Sargent, Gehre-Hallwachs, Storrer, Eckardt, and Venturi steam meters the velocity is increased by throttling and the pressure drop is utilized in actuating the mechanism. The weight of steam flowing through the orifice may be calculated from the following modification of equations (286) and (287):

$$W = AK\sqrt{p_1 - p_2}, \tag{290}$$

in which

W = pounds discharged per second.

A = area of the orifice, square feet.

K = coefficient determined by experiment and includes the density of the steam.

p_1 and p_2 = pressure on the upper and lower side of the orifice, pounds per square inch.

In some of the meters the pressure drop $p_1 - p_2$ is maintained constant and the variation in the area A actuates the indicating mechanism, and in others the area is made constant and the variation in pressure drop operates the mechanism.

Fig. 562 represents a section through a St. Johns steam meter, illustrating the throttling type with a floating valve. This meter was placed on the market 20 years ago and still finds favor with many engineers. It records the weight of steam passing through the seat of an automatically lifting valve which rises and falls as the demand for steam increases or diminishes.

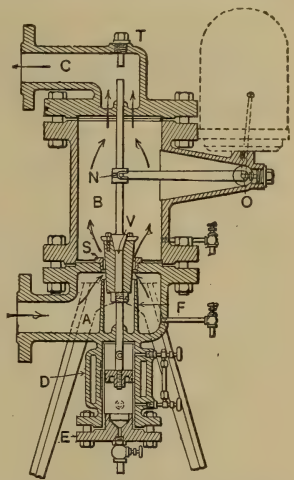


Fig. 562. St. Johns Steam Meter.

Referring to the illustration, valve V is weighted so that a pressure in space A of 2 pounds greater than in B is necessary to raise the valve off its seat. This pressure difference is constant for all positions of the valve. The plug is tapered so that the rise of the steam pressure is directly proportional to the volume of steam flowing through the seat. The movement of the valve is transmitted through suitable levers to an indicating dial and a recording pen so that the instantaneous and continuous rate of flow may be read at a glance. For a given pressure

Fig. 563. Different Forms of Manometer Pressure Gauges.

and quality of steam, the indicating dial and chart may be calibrated to read the weight of discharge directly, corrections being made for variations in pressure and quality. The manufacturers guarantee the readings of the chart to be within 2 per cent of condenser measurements for a total pressure range of 10 pounds from the mean pressure at which the chart is calibrated.

The chief drawback to this instrument is inherent to all meters of the direct type in that they are bulky and the steam line must be taken down for the installation. The total hourly flow may be obtained by integrating the curve. Tests of this meter made by the author were in accordance with the guarantee of the manufacturer for continuous flow and for moderate changes in the rate of flow. For rapid fluctuations in flow the results were not so satisfactory, the greater error lying in the difficulty of integrating the curve correctly.

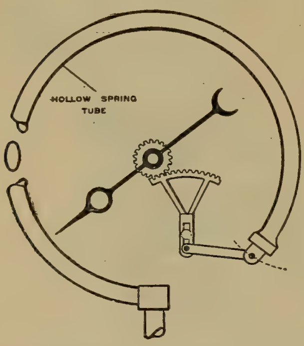


FIG. 564. Bourdon Pressure Gauge.

445. Pressure Gauges. — The Bourdon type of gauge, either autographic or indicating (Fig. 564), is the most familiar and satisfactory means of measuring pressures up to 1500 pounds per square inch or more, although diaphragm gauges are also used and both are employed as vacuum gauges. For the latter purpose, however, the mercurial vacuum gauge has the advantage of greater accuracy and is not subject to derangement. Bourdon gauges should be frequently standardized by comparison with a gauge of known accuracy, a mercury column, or a gauge tester.

For measuring very low pressures, such as are found in boiler flues or gas mains, indicating or recording diaphragm gauges may be had, but some form of U-tube manometer is generally employed, the design best adapted to the purpose depending upon the accuracy required. The simple U-tube (Fig. 563) when filled with mercury may be used for pressures limited only by the inconvenience due to length of tubes, or with water as the fluid, for pressures only a fraction of an ounce per square inch. Where greater accuracy is required than can be obtained with the simple U-tube, some modification may be employed, such as the Ellison draft gauge with one inclined leg which magnifies the reading several times. A form of sensitive gauge is sometimes used which depends upon the use of two fluids of different specific gravity, as oil and water.

Pressure Gauges: Power, Aug. 15, 1911, p. 239; Aug. 18, 1908, p. 286.

446. Measurement of Temperature. — For power-plant purposes mercurial thermometers are most convenient for measuring temperatures up to 400 degrees F., and are inexpensive. For higher temperature, up to say 800 degrees F., they are also adapted, but must be made of special glass and the space above the mercury filled with nitrogen under pressure to prevent vaporization of the mercury. Such thermometers

must be used intelligently and should be standardized from time to time, since they are subject to considerable change. The Bureau of Standards at Washington, D.C., is prepared to furnish certificates for which a nominal charge is made.

Fig. 565 shows a form of thermometer which is much used where a continuous autographic record is required. It depends for its operation upon the pressure produced by a fluid, liquid or gaseous, contained in a small bulb and exposed to the temperature to be measured. The pressure is transmitted to the recording mechanism through a flexible

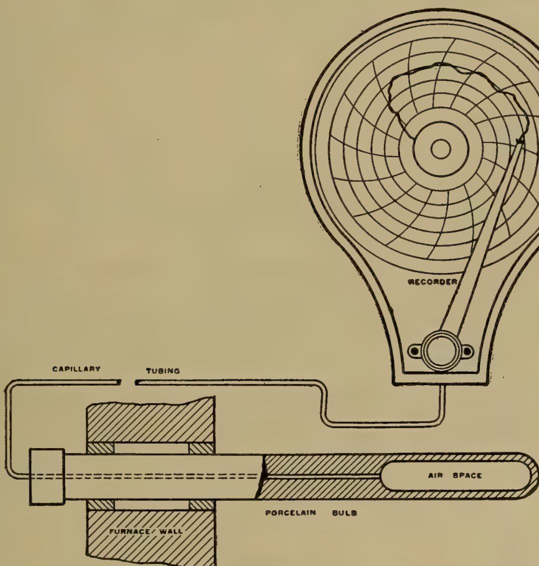


FIG. 565. Bristol Recording Pyrometer.

capillary tube which may be of considerable length. Such thermometers are suitable for feed water, flue gas, and temperatures not exceeding 1000 degrees F.

Fig. 566 illustrates a form of electrical pyrometer employing thermocouples which has come into wide use as a reliable means of measuring temperatures up to 2600 degrees F. The couples most frequently used are composed of platinum and platinum-rhodium, platinum and platinum-iridium, copper and copper-constantan, and copper and nickel, the first named being adapted to the higher ranges of temperature. The electromotive force set up, when the thermo-junction is heated, is proportional to the temperature and is measured by means of a sensitive millivoltmeter which is usually graduated to read temperature directly.

Thermo-couples may be made to give an autographic record by means of a *thread recorder.*

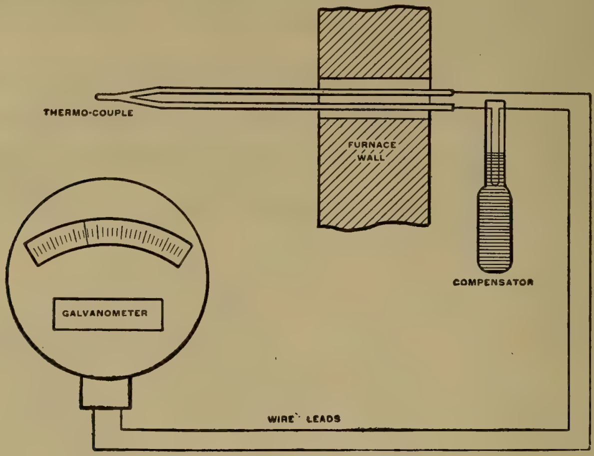


FIG. 566. Bristol Thermo-Electric Pyrometer.

Fig. 567 shows the element of an electrical thermometer based upon the change in resistance of a platinum wire when subjected to change in temperature. The resistance, in terms of temperature, is measured by a *Whipple indicator,* a convenient and portable form of Wheatstone bridge, or may be autographically recorded by means of a *Callendar recorder.* Resistance thermometers of this type are very sensitive and accurate, not easily deranged, and are limited in range only by the fusing points of the platinum and the porcelain protecting sheath.

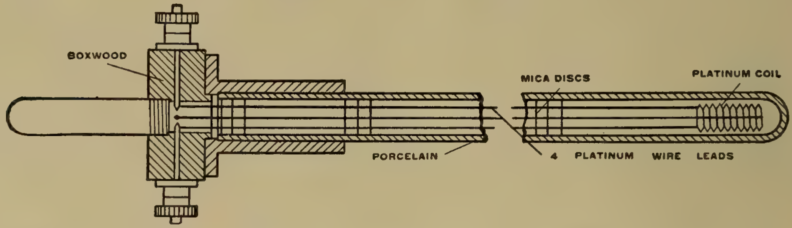


FIG. 567. Element for Callendar Resistance Pyrometer.

For higher temperatures and for obtaining the temperatures of inclosed spaces above about 900 degrees F., such as boiler furnaces, annealing ovens, and kilns, various forms of *optical* and *radiation pyrometers* have been devised. In such devices no part of the instrument is exposed to the temperature to be measured and hence suffers no injury from this cause. Optical pyrometers are based upon the measurement of the brightness of the hot body by comparison with a

standard. The Wanner optical pyrometer is shown in Fig. 568. After standardizing by comparison with an amyl-acetate lamp, it is only necessary to focus the instrument upon the source of heat to be measured and the temperature is read on the graduated scale.

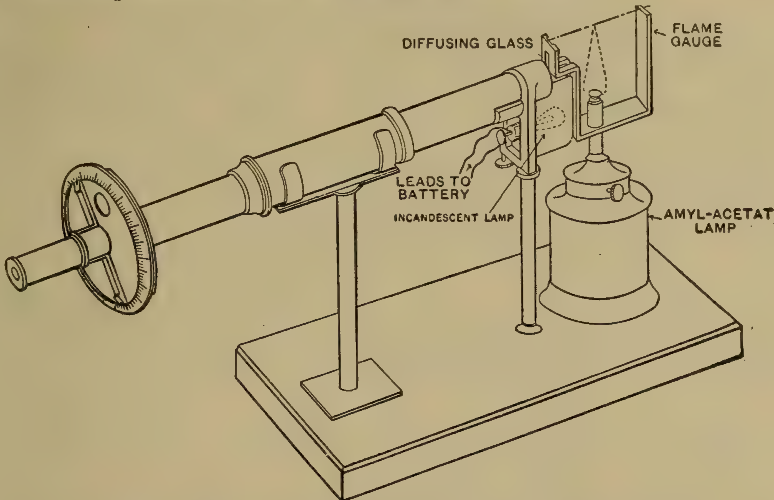


Fig. 568. Wanner Optical Pyrometer in Position for Standardizing.

Radiation pyrometers depend upon the measurement of the heat radiated from the hot body. The Féry radiation pyrometer, Fig. 569, is the best-known instrument of this type. When focused upon the

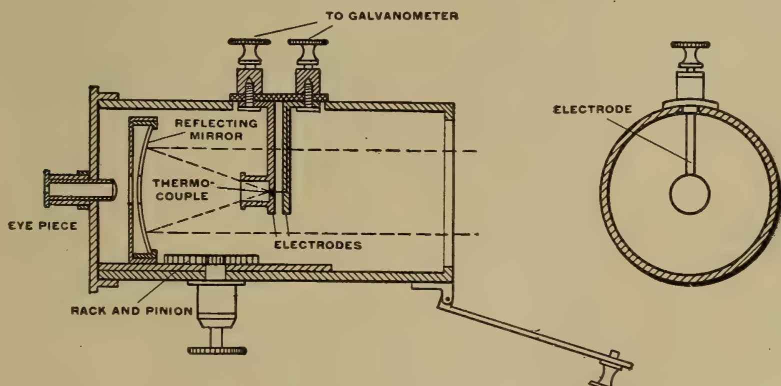


Fig. 569. Féry Radiation Pyrometer.

source of heat a cone of rays of definite angle is reflected by means of the mirror upon a thermo-couple located in its focus. The electromotive force set up is measured in terms of the temperature of the source of heat by a millivoltmeter. Neither the couple nor any part of the instrument

is ever subjected to a temperature much above 150 degrees F. The indications are practically independent of the distance from the source of heat, and the range is without limit.

TABLE 127.

TYPES OF THERMOMETERS IN GENERAL USE.

Principle of Operation.		Type.	Range in Degrees F. for which they can be used.
Expansion	Those depending on the change in volume or length of a body with temperature.	Gas	−400 to +2900
		Mercury, Jena glass, and nitrogen.	−35 to +950
		Glass and petrol ether.	−325 to +100
		Unequal expansion of metal rods.	0 to 950
Transpiration and viscosity.	Those depending on the flow of gases through capillary tubes or small apertures.	The Uehling	0 to 2900
Thermo-electric	Those depending on the electro-motive force developed by the difference in temperature of two similar thermo-electric junctions opposed to one another.	Galvanometric	−400 to +2900
Electric resistance	Those utilizing the increase in electric resistance of a wire with temperature.	Direct reading on indicator or bridge and galvanometer.	−400 to +2200
Radiation	Those depending on the heat radiated by hot bodies.	Thermo-couple in focus of mirror.	300 to 4000
		Bolometer	−400 to Sun
Optical	Those utilizing the change in the brightness or in the wave length of the light emitted by an incandescent body.	Photometric comparison. Incandescent filament in telescope. Nicol with quartz plate and analyzer.	1100 to Sun
Calorimetric	Those depending on the specific heat of a body raised to a high temperature.	Platinum ball with water vessel.	32 to 3000
Fusion	Those depending on the unequal fusibility of various metals or earthenware blocks of varied composition.	Alloys of various fusibilities. (Seger cones.)	32 to 3350

The *Uehling pyrometer* depends for its operation upon the flow of gas between two apertures, thus: Air is continuously drawn through two apertures by a constant suction produced by an aspirator. So long as the air has the same temperature in passing through these orifices there

is no change in the partial vacuum in the chamber between them; if, however, the air passing through the first opening has a higher temperature than that passing through the second, the vacuum in the chamber will increase in proportion to the difference in temperature since the volume of air varies directly with the temperature. In the application of this principle, the first aperture is located in a nickel tube which is exposed to the heat to be measured, while the second aperture is kept at a uniform lower temperature. This style of pyrometer is made to indicate and record and the indicating and recording mechanism can be placed at a distance from the main instrument.

Table 127 embodies in outline the principles and temperature ranges of the various types of thermometers in use. Temperature ranges verified by U. S. Bureau of Standards.

Modern Methods of Temperature Measurements: Cassier's Mag., June, 1909, p. 99. *High Temperature Measurements:* Eng. and Min. Jour., Sept. 2, 1911, p. 447; Power, Aug. 2, 1910, p. 1376; Engineering, Feb. 9, 1912, Bul. No. 2, Bureau of Standards.

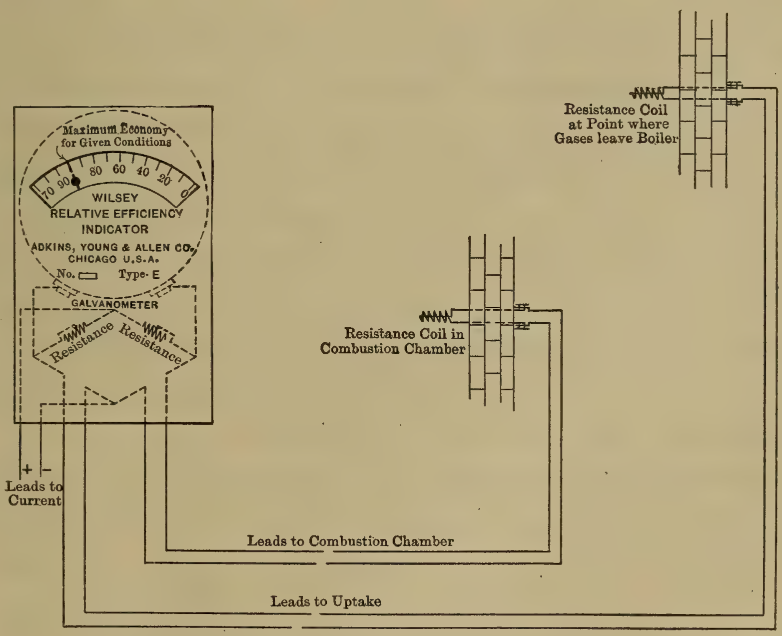


FIG. 570. Wilsey Relative Efficiency Indicator.

447. Wilsey Relative Efficiency Indicator. — Fig. 570 shows the general principles of the Wilsey relative efficiency indicator for indicating the relative boiler and furnace efficiency in any given installation. It consists essentially of two platinum resistance coils, a modified

Wheatstone bridge, and a millivoltmeter. One of the coils is placed in the combustion chamber, the other in the uptake, and the indicating mechanism is mounted on the front of the boiler in plain view of the fireman. The ratio between the temperature of the combustion chamber and the uptake is a function of the boiler and furnace efficiency. The indicator is calibrated for each installation, so that when the boiler and furnace are operated at maximum efficiency under the given conditions, the indicating needle points to "maximum efficiency" or 100 per cent. If excess air is admitted the needle shows a lower relative efficiency due to the reduction in temperature, and if the air supply is deficient the needle swings beyond the predetermined "maximum efficiency." This device considers the efficiency of combustion solely from a qualitative standpoint regardless of capacity. When installed in connection with a boiler flow meter the fireman is able to note the capacity and relative efficiency at a glance. Tests of this apparatus at the boiler plant of the Armour Institute of Technology gave reliable and consistent results.

For a description of the Blonck "Boiler-Efficiency" meter see Elec. Rev. and Wes. Elec., Oct. 12, 1912; Prac. Engr. U. S., Feb. 15, 1913.

448. Power Measurements.—Instruments for the measurement of power may be divided into two general classes, *direct* and *indirect*. The former involve the direct measurement of force and linear velocity or torque and angular velocity and the latter give the equivalent in other forms of energy. Direct power measuring appliances include the various speed indicators, transmission and absorption dynamometers, and the indirect include ammeters, voltmeters, watt-hour meters, boiler flow meters, and the like. In all power measurements the time or speed factor is readily determined but the force or torque factor, or equivalent, often involves considerable labor and the use of costly and complicated apparatus. The various conversion factors for the measurement of work, power, and duty are given in Appendix J.

449. Measurement of Speed. — The following chart gives a classification of a number of well-known instruments for determining linear and angular velocities.

Counters	Hand	Worm and Wheel.
	Continuous	Gear Train. Electrical.
Tachometer or Speed Indicators	Centrifugal	Weights. Liquids.
	Electrical.	
	Resonance	Frahm's.
Chronograph	Electro-magnetic Tuning Fork.	

The most commonly used device for speed determinations is the *hand speed counter*, consisting of a worm, worm wheel, and indicating dials. The errors to be corrected are principally those due to slipping of the point on the shaft, and to the slip of the gears in the counting device in putting in and out of operation. In some of the better grade of instruments the gears are engaged or disengaged with the point in contact with the shaft. In the latter design a stop watch, actuated by the disengagement gear, minimizes the error likely to occur in hand manipulation.

The *continuous counter* consists of a series of gears arranged to operate a set of indicating dials. It may be operated by either rotary or reciprocating motion. The rate of rotation is calculated from the readings of the counter.

All *tachometers* indicate directly the speed of the machine to which they are attached and are independent of time determination. The most commonly used devices depend upon the centrifugal force of revolving weights for their operation. The indicating needle is attached to the weights in such a manner that the number of revolutions per minute is read directly from the position of the needle on the dial. These instruments should be calibrated for accurate work because of the number of wearing parts.

Liquid tachometers consist essentially of small centrifugal pumps discharging into a vertical tube. The height of the indicating column is a function of the speed of rotation.

Electrical tachometers are miniature dynamos, the voltage being a measure of the speed of rotation. These instruments are accurate and readily attached but necessitate the use of a delicate and costly voltmeter. The indicating mechanism may be placed at any distance from the small dynamo and in this respect has a marked advantage over the other types of speed indicators.

The *resonance tachometer* affords a convenient method of measuring speeds over a wide range. It consists of a number of steel reeds of different periodicity mounted side by side on a suitable frame. When used to measure the speed of an engine or turbine the instrument is placed on or near the bed plates and the slight under or over balance causes the proper reed to vibrate in unison.

450. Steam-engine Indicators. — This subject has been extensively treated by various authorities and a general discussion would be without purpose. For *indicated horse power*, *testing indicator springs*, and analysis or indicator diagrams see articles XII, XIV, and XX, "Rules for Conducting Steam Engine Tests," A.S.M.E., code of 1898. See also Preliminary Code for 1912, Jour. A.S.M.E., Nov., 1912.

451. Dynamometers. — Dynamometers for measuring power are of two distinct types, *absorption* and *transmission*. In the former the power is absorbed or converted into energy of another form while in the latter the power is transmitted through the apparatus without loss, except for minor friction losses in the mechanism itself.

The ordinary *Prony brake* is the most common form of absorption dynamometer. In the various forms of Prony brakes the power is absorbed by a friction brake applied to the rim of a pulley. For low rubbing speeds and comparatively small powers it affords a simple and inexpensive means of measuring the actual output.

The Alden absorption dynamometer is a successful form of friction brake and has a wide field of application. It has been constructed in large sizes and is adapted to all practical ranges of speed. For a description of rope brakes and the Alden absorption dynamometers see Appendix No. 19, p. 1848, Jour. A.S.M.E., Nov., 1912.

Water brakes are finding much favor with engineers for high-speed service. There are two types, the Westinghouse and the Stumpf. In the former the rotor consists of a simple drum with serrated periphery revolving in a simple casing, the inner surface of which is serrated in a manner similar to the rotor. The resistance is produced by friction and impact, and the power is converted into heat which is carried away by the circulating water. The casing is free to turn about the shaft but is held against rotation by a lever arm. The torque of the lever arm is determined as in a Prony brake. A brake of this design, 2 feet in diameter and 10 inches wide will absorb about 3000 horse power at 3500 r.p.m. In the Stumpf type the rotor consists of a number of smooth disks mounted side by side on a common shaft. The casing is divided into a number of compartments corresponding to the division of the rotor. There is no contact between rotor and casing. The friction between the disks and water and the water and casing tends to rotate the latter and the torque is measured in the usual way. In either type the power output is readily controlled by the water supply.

Pump brakes and *fan brakes* are also used as absorption dynamometers. The latter are commonly used in connection with automobile engine testing.

Electromagnetic brakes are occasionally used for power measurements. They consist essentially of a metal disk or wheel revolving in a magnetic field. The resistance or drag tends to revolve the field casing and the torque is measured in the usual way.

An electric generator mounted on knife edges forms the basis of the Sprague electric dynamometer. The prime mover drives the armature of the generator and the reaction between armature and field is counter-

balanced by suitable weights. The output is conveniently regulated by a water rheostat.

Transmission dynamometers are seldom used for testing prime movers and are ordinarily limited to small power measurements. In some instances, however, as in marine service, transmission dynamometers afford the only practical means of approximating the net power delivered to the propeller. For comparatively small power measurements may be mentioned the Morin, Kennerson, Durand, Lewis, Webber and Emerson transmission dynamometers, and for large powers, the Denny and Johnson electrical torsion meter and the Hopkinson optical torsion meter. For detailed descriptions of these appliances consult "Experimental Engineering," Carpenter and Diederichs, Chap. X.

452. Flue Gas Analysis. — It has been shown (paragraph 20) that the products of combustion, commonly called flue gases, resulting from the complete oxidation of coal with theoretical air supply consist chiefly of nitrogen and carbon dioxide, with lesser amounts of water vapor and sulphur dioxide. It was also shown that with a deficient air supply the flue gases may contain carbon monoxide and varying amounts of hydrocarbon. If excess air was used in the combustion of the fuel free oxygen would be present in the gases. Evidently an analysis of the flue gases offers a basis for judging the efficiency of combustion. The first step in the analysis and the most important one is the obtaining of a representative sample. Since the gases in the breeching and flues may be far from homogeneous great care must be exercised in getting a true average sample. (See *Apparatus and Methods for Sampling and Analysis of Furnace Gases*, U. S. Bureau of Mines, Bul. No. 12, 1911.)

The analysis as ordinarily made in commercial practice is called volumetric, although in reality it is based upon the determination of partial pressures. According to Dalton's laws (paragraph 239) when a number of gases are confined in a given space each gas occupies the total volume at its own partial pressure, and the total pressure is the sum of all the partial pressures. When one of the gases is absorbed by a suitable medium and the remaining gases are compressed back to the original total pressure, a volume decrease is found, and if the temperature remains constant this decrease represents the volume absorbed.

The apparatus usually employed for volumetric analysis consists of a graduated measuring tube into which the gases are drawn and accurately measured under a given pressure, and a series of treating tubes, containing the necessary absorbing reagents, into which they are transferred until absorption is complete. The *Orsat apparatus*, Fig. 571, forms the basis of nearly all of the portable appliances on the market for analyzing flue gases and the ordinary products of combustion. In

this apparatus a measured volume, representing an average sample of the gas, is forced successively through pipettes containing solutions of caustic potash, pyrogalic acid and cuprous chloride in hydrochloric acid, respectively, thus absorbing the carbon dioxide, the oxygen and the carbon monoxide, the contraction of volume being measured in each case. The apparatus as originally constructed is bulky and fragile and slow in its absorption of gas.

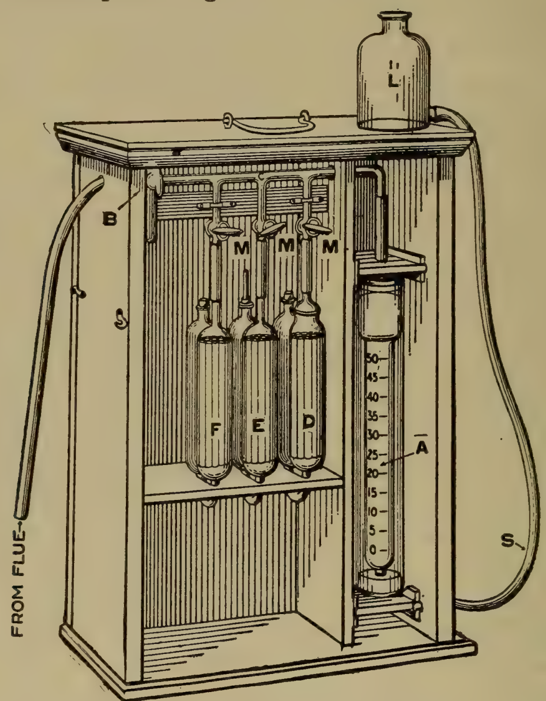


FIG. 571. Standard Orsat Apparatus for Flue Gas Analysis.

453. The Williams Improved Gas Apparatus is a marked improvement over the standard Orsat in that the objections cited above are obviated. In addition to the elimination of these objectionable features provision is made in the "Model A" type for the determination of illuminants, hydrogen and methane along with the three gases mentioned above. Referring to Fig. 572, *A*, *B*, *C*, and *D* are pipettes containing the necessary reagents for absorbing, respectively, CO_2, illuminants, O_2, and CO. *M* is a graduated measuring flask connected at the bottom with water-level bottle *W* and at the top with the various pipettes. *F* is a hard rubber pump for taking gas sample directly from the source of supply, thereby eliminating the inaccuracy and annoyance of collection over water and transference. *P* is a portable case containing a spark

coil and batteries for exploding the methane and hydrogen remaining in the burette after the other constituents have been removed. When extreme accuracy is desired mercury is used as the displacement medium in the leveling bottle since water absorbs CO_2 to a certain extent. For a complete description of this apparatus with sample calculations see paper read by F. M. Williams before Division of Industrial Chemists and Chemical Engineers, American Chemical Society, Washington, D. C., Dec. 28, 1911.

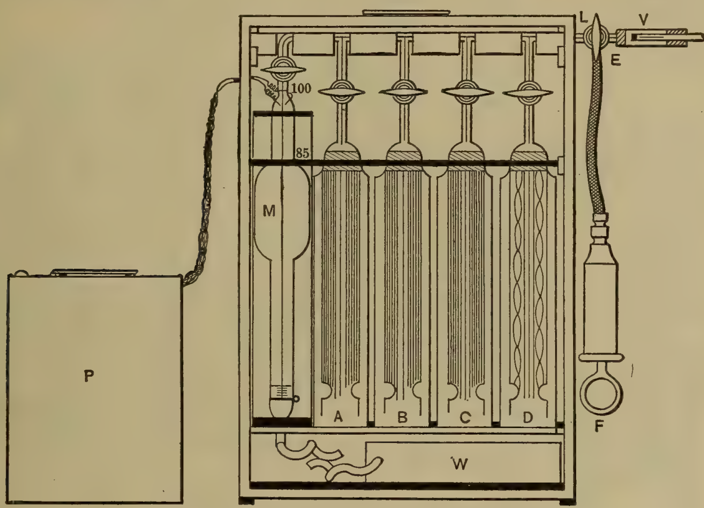


Fig. 572. Williams Improved Gas Apparatus.

454. "Little" Modified Orsat Apparatus.—Fig. 573 illustrates a modified Orsat apparatus as used by the Arthur D. Little Company of Boston, Mass. The right half of the device is the ordinary Orsat apparatus and the left portion constitutes the sampling attachment. The gases are drawn from the source of supply through rubber tube (2) into the sampling pipette (3) and out through rubber tube (1) to the aspirator. The latter may be operated by steam or water. When a sample is being collected the three-way cock on the glass header is closed and the mercury in the sampling tube (4) is allowed to drain through the movable overflow into the mercury retainer. The overflow is lowered at a constant rate by clockwork. Two driving pulleys afford seven different rates of movement downward of the overflow, thereby enabling a continuous sample to be collected at constant rate over any period from $\frac{1}{2}$ to 24 hours. Instantaneous samples may be drawn off and analyzed as often as desired and with practically no delay to the continuous sample. For further details see Power, July 16, 1912, p. 77.

For many practical purposes it is sufficient to determine the carbon dioxide. A number of satisfactory appliances are on the market which give continuous autographic records of the percentage of CO_2 on clock-driven charts. These devices, however, are rather expensive and usually beyond the appropriation of small boiler plants.

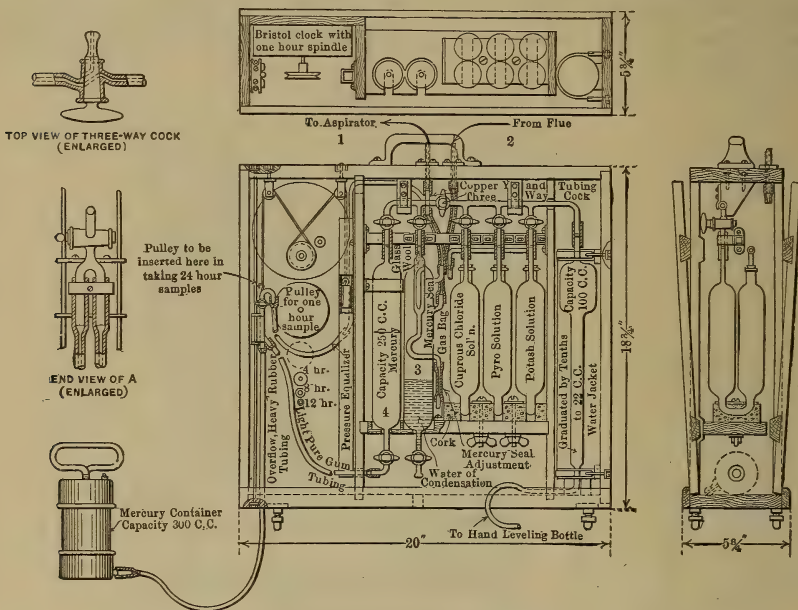


FIG. 573. Modified Orsat Apparatus. — Arthur D. Little Co.

455. Simmance-Abady CO_2 Recorder. — Fig. 574 illustrates the general principles of the *Simmance-Abady CO_2 Recorder.* The operation is as follows: A continuous stream of water enters reservoir K through inlet X and overflow at O. A portion of the stream flows into tank A through pipe F and causes bell float B to rise. As the float rises it permits bell D of the extractor to fall. When float B reaches the top of its stroke it raises valve stem E, trips the valve and causes the water to siphon out of tank A through siphon tube G. The lowering of the water level allows the bell to sink. As it falls it draws up the water-sealed extractor bell D and creates a partial vacuum under the latter. Flue gas then flows from the source of supply through P and H into the bell. The mass of water discharged from siphon tube G into the small vessel V beneath it overcomes the counterweight Q and closes the balance valve H, thereby entrapping a fixed volume of gas in the extractor bell. The stream of water which is continually flowing into tank A causes the float B to rise and the bell D to sink, as

before. The lowering of bell D forces the entrapped flue gas through the caustic potash solution in vessel M into water-sealed recorder bell J. The displacement of bell J will be less than that of bell D by the volume of CO_2 absorbed in vessel M. The percentage of CO_2 in the flue gas is thus indicated by the position of the bell J with reference to the graduated scale N. The pen mechanism is attached to bell J

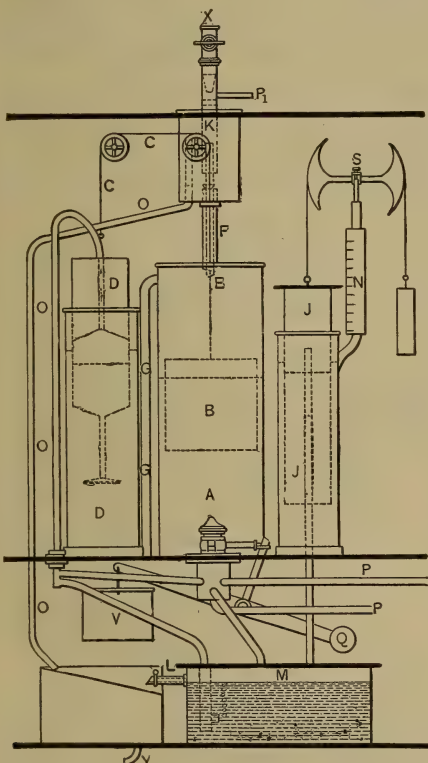


FIG. 574. Simmance-Abady CO_2 Recorder.

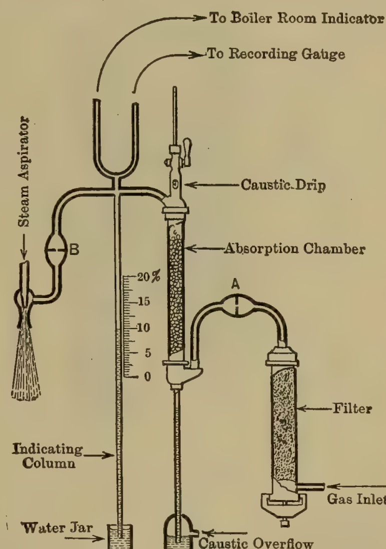


FIG. 575. Principles of the Uehling Gas Composimeter.

and records the percentage of CO_2 by the length of lines on a clock-driven chart. These samples are analyzed and the lines are drawn at three-minute intervals. The small water aspirator at X is for the purpose of exhausting gas continuously from the pipes connecting the recorder to the boiler, thereby insuring true samples at the time of absorption. Auxiliary pipe P is connected to main gas lead P.

456. The Uehling Composimeter is another successful instrument for continuously recording the percentage of CO_2 in the flue gas. The principles of this apparatus are illustrated in Fig. 575. The device consists primarily of a filter, absorption chamber, two orifices, A and

B, and a small steam aspirator. Gas is drawn from the usual source by means of the aspirator through a preliminary filter located at the boiler, and then through a second filter as illustrated in the diagram. From the latter the gas passes through orifice *A*, thence through the absorption chamber and orifice *B* to the aspirator where it is discharged. The CO_2 is absorbed by the caustic potash solution in the absorption chamber. This reduces the volume and causes a change in tension between the two orifices in proportion to the CO_2 content of the gas. This variation in tension is indicated by the water column, as shown, and is transmitted by suitable piping to the recording mechanism which may be placed at a considerable distance from the boiler room.

457. Moisture in Steam. — Several forms of calorimeters are available for determining the quality of steam. The simplest as well as

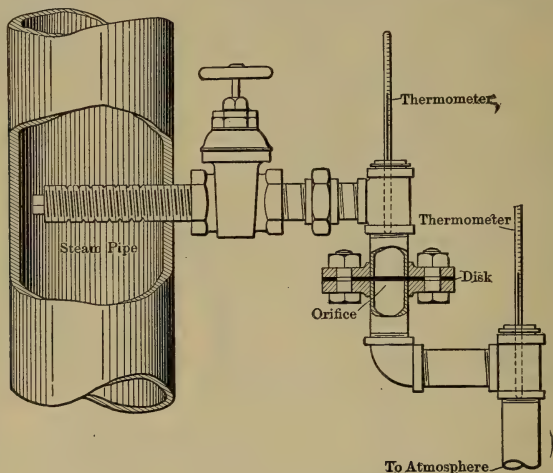


Fig. 576. A Typical Throttling Calorimeter.

the most satisfactory, if the percentage of entrained moisture is not beyond its range, is the *throttling* calorimeter, Fig. 576. In this device the sample of steam, which is taken from the steam pipe by means of the perforated nipple, is allowed to expand through a very small orifice into a chamber open to the atmosphere. The excess of heat liberated serves first to evaporate any moisture present and then to superheat the steam at the lower pressure. From the observed temperature and pressures it is easy to calculate, with the aid of steam tables, the percentage of moisture in the original sample.

The limit of the throttle calorimeter depends upon the steam pressure and is about 3 per cent of moisture at 80 pounds pressure and about 5 per cent at 200 pounds. For steam containing greater percentages of moisture the *separating* calorimeter, Fig. 577, is sometimes used.

This instrument is virtually a steam separator and mechanically separates the moisture from the sample of steam. The water thus separated collects in a reservoir provided with gauge glass and graduated scale, while the dry steam passes through an orifice to the atmosphere. The weight of dry steam per unit of time is indicated on the gauge, calculated according to Napier's rule, or may be determined by condensing and weighing. The accuracy of the moisture determination is greatly affected by the difficulty of obtaining true samples of steam containing large percentages of moisture.

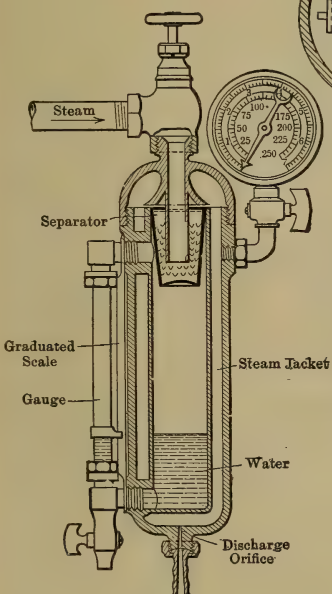


FIG. 577. Carpenter Separating Calorimeter.

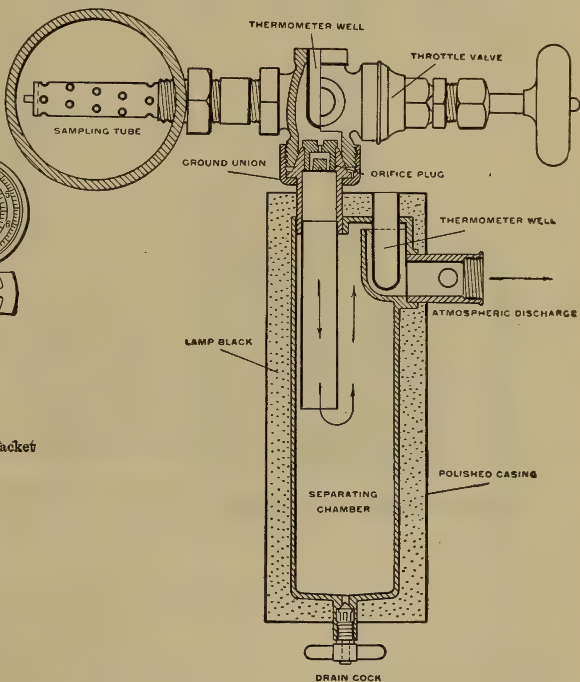


FIG. 578. Ellison Universal Steam Calorimeter.

Fig. 578 shows the Ellison *universal* steam calorimeter, which combines the superheating and throttling principles and is adapted to steam of any degree of wetness. The separating chamber is provided with a gauge glass, not shown, for indicating the weight of water which accumulates only when the steam is too wet to be superheated.

Throttling Calorimeters: Power, Dec., 1907, p. 891; Trans. A.S.M.E., 17–151; 175, 16–448; Engr. U. S., Feb. 15, 1907, p. 219.

Separating Calorimeters: Trans. A.S.M.E., 17–608; Engr. U.S., Feb. 15, 1907, p. 219.

Universal Calorimeter: Trans. A.S.M.E., 11–790.

Thomas Electrical Calorimeter: Power, Nov., 1907, p. 791.

458. Fuel Calorimeters. — The analysis and heat evaluation of fuel require considerable time and skill and much costly apparatus, hence in most power plants it is customary to depend upon a specialist to whom samples are submitted from time to time. In many large stations, however, the conditions often warrant the establishment of a testing laboratory equipped for the proximate analysis of coal and the determination of the calorific value of the solid, liquid or gaseous fuel used. The *Mahler bomb* calorimeter illustrated in Fig. 579 is the most accurate and satisfactory device for solid and liquid fuels but is comparatively expensive. The instrument consists of a steel shell or

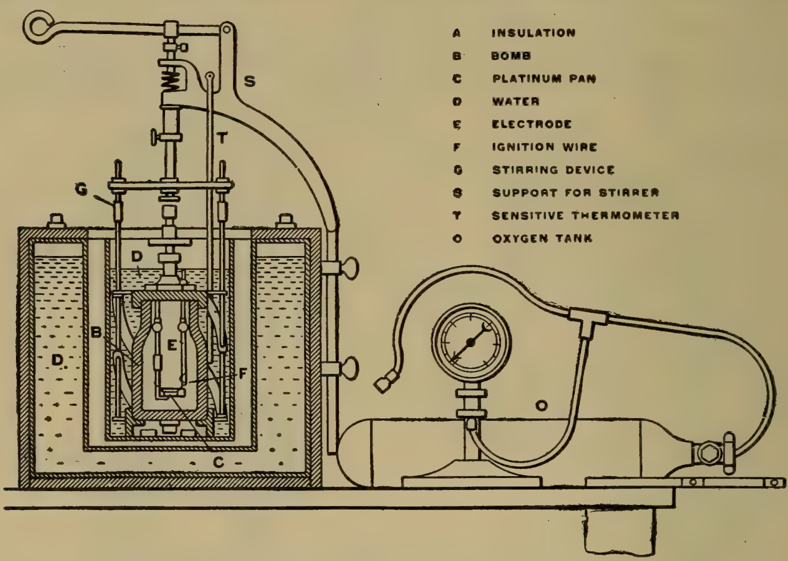


Fig. 579. Mahler Bomb Calorimeter.

"bomb" of great strength, lined with porcelain or platinum, into which a weighed sample of the fuel is introduced and burned on a platinum pan in the presence of oxygen under a pressure of about 300 pounds per square inch. The charge is ignited by an electric current. During combustion the bomb is submerged in a known weight of water which is kept constantly agitated. The calorific value is calculated from the observed rise in temperature due to the heat evolved, proper corrections being made for the water equivalent of bomb and appurtenances, heat given up by the igniting current, and for radiation or absorption of heat from the surrounding air.

The *Parr* calorimeter, Fig. 580, is an inexpensive instrument, very simple in operation, and gives results which are sufficiently accurate for all practical purposes. The weighed sample of coal, together with

a quantity of sodium peroxide which supplies the oxygen for combustion, is introduced into the cartridge. Means are provided for rotating the cartridge when submerged in the calorimeter, the attached vanes agitating the water to maintain uniform temperature.

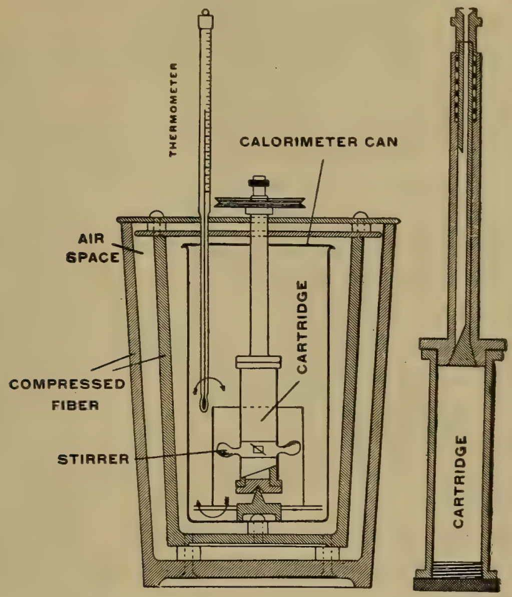


Fig. 580. Parr Fuel Calorimeter.

The charge is fired either electrically or by introducing a short piece of hot wire through the conical valve. The calorific value is calculated from the observed rise in temperature and the constants of the instrument. Among other forms of instruments, in more or less general use and which give very satisfactory results, may be mentioned the *Carpenter, Thompson, Atwater* and *Emerson* calorimeters.

Comparison of Different Types of Calorimeters: Jour. Soc. Chem. Ind. (1903), 22–1230.

CHAPTER XVIII.

FINANCE AND ECONOMICS.—COST OF POWER.

459. General Records. — In all power plants, public or private, an itemized record of plant performance and cost of operation is of vital importance for the most economic results. In many states public utility corporations are required to submit an annual statement covering the various details of operation, and in order to insure uniformity ruled and printed forms are furnished by the state. The private plant owner, on the other hand, is free to use his own judgment and may adopt any system of cost accounting or dispense with them entirely.

The principal objects of keeping a system of records are (1) to enable the owner to compare the performance of his plant with current practice and to show him exactly what his plant is costing him, and (2) to enable the engineer to analyze the various records with a view of reducing all losses to a minimum. Power-plant records to be of value must be closely studied with a view to improvements. The mere accumulation of data to be filed away and never again referred to is a waste of time and money.

Records should cover not only the daily, monthly, and yearly operation of the plant but also, as permanent statistics, a complete analysis of each item of equipment. The value of such data cannot be overestimated. The engineer will frequently find it greatly to his interest to have available at a moment's notice the complete details of his engines, boilers, generators, and other machinery, especially when it is required to renew a broken or worn-out part in case of emergency.

The question of whether to purchase power or to generate it depends, chiefly, upon the relative cost of the two methods, although the absence of power-plant machinery and freedom from the coal and ash handling nuisance may be important factors. There is no doubt but that the central station can generate power cheaper than the small isolated plant, but in most cases it is a question not only of power, but also of supplying steam for heating and other purposes, and a careful study of all of the items entering into the problem is necessary for an intelligent choice. The service department of the large central station with its carefully maintained system of records has a strong advantage in presenting its arguments over the average private plant with its ill-kept

and faulty system of accounting, and in some instances central-station service has been adopted simply because the engineer in charge was not in a position to prove positively that his own plant was the better investment.

TABLE 128.

PERMANENT STATISTICS.

General Information.

Date of installation	....	Total cost of building	$5,000,000
Type of building	Office	Ground plan	191×231
Number of floors	18	Total office floor space, sq. ft.	400,000
Number of offices	900	Height of building	280
Volume of building, cu. ft.	10,860,000	No. of sides exposed	3
Type of heating system	Webster	Radiator surface, sq. ft.	100,000
Engine room, sq. ft.	6,840	Boiler room, sq. ft.	5,400
Height of chimney, ft.	318	Number of elevators	22
Draft, inches of water	3.5	Type of elevators	High pressure hydraulic
Kind of grate or stoker	Jones Underfeed	Capacity of elevators, lb., each	2,700
Kind of coal	Ill. screenings	Boiler pressure	150
Coal storage capacity, tons	450	Back pressure	Atmospheric
Capacity ice plant, tons in 24 hrs.	50	Part of bldg. lighted	All
Capacity storage battery, am. hrs.	None	Total cost of mechanical plant	$650,000

	Engines.	Generators.	Motors.	Boilers.
Type	Ball compd.	Crocker-Wheeler		
Number installed	5	5	25	5
Rated capacity	250 h.p.	150 kw.		375 h.p.

LIGHTS.

	Incandescent.		Arcs.
Type	Carbon	Tungsten	Inclosed
Number installed	25,000	5000	15

A number of attempts have been made to standardize power-plant records but the results have been far from satisfactory because of the wide range in operating conditions. Each installation is a problem in itself and the items to be recorded must necessarily depend upon the size and character of the plant. A common mistake is to attempt too comprehensive a system with the result that after the novelty has ceased the labor of making the various entries becomes irksome, many of the items are omitted, guesses are substituted in place of actual observations, and the records are ultimately without value. A few

properly selected items, accurately recorded, are of vastly more importance than an elaborate system of records indifferently maintained.

460. Permanent Statistics. — Tables 128 to 131 are taken from the records of a large isolated station in Chicago and serve to illustrate the make-up of the "permanent statistics." The complete file covers each item of equipment and includes the various drawings, specifications, and guarantees for the entire mechanical equipment. Since these records do not vary with the operation of the plant they require no further attention, once they are compiled, except of course for such changes as may be made from time to time in the plant itself.

TABLE 129.

PERMANENT STATISTICS.

BOILERS.

Make of boiler	Stirling	Number in battery	1
Total number in plant	5	Weight of boiler	62,186
Date of installation	....	Cost of boiler and fittings (each)	$5,400
Steam pressure, gauge	150	Height of setting	17 ft. 9 in.
Safety-valve pressure	160	Length of setting	17 ft. 4 in.
Type of safety valve	Pop	Width of setting	15 ft. 3 in.
Area of grate, sq. ft.	....	Weight of setting	272,000
Heating surface, sq. ft.	3,500	Thickness of wall	Side 20 in.; back, 15 in.
Superheating surface, sq. ft.	None	No. of bricks, fire	6,590
Number of steam drums	3	No. of bricks, common	19,600
Diameter of steam drums, in.	36	Dimensions of foundation	15 ft. 2 in. × 17 ft. 4 in.
Distance between steam drums, ft.	3	Material of foundation	Stone and concrete
Thickness of shell, in.	$\frac{3}{4}$	Cost of foundation and setting (each)	$1,500
Thickness of head, in.	$\frac{3}{4}$	Distance between batteries	4 ft. 6 in.
Diameter of steam nozzle, in.	10	Distance back of boiler	17 ft. 6 in.
Diameter of safety valve	2–4 in.	Distance in front of boiler	16 ft. 6 in.
Diameter of blow-off, in.	2.5	Distance overhead	2 ft. 10 in.
Diameter of feed pipe, in.	2	Number of tubes	337
Temperature of flue, deg. Fah.	450–490	Diameter of tubes, in.	3.25
Temperature of feed water, deg. Fah.	210	Length of tubes, ft.	12 to 14
Ratio of heating surface to grate area	41.6	Steam space, cu. ft.	96
Kind of fuel	Carterville, Ill., Screenings	Water space, cu. ft.	643
Type of grate	Green chain grate	Kind of draft	Forced
Rated horse power	375	Inches of draft (maximum)	3.5

461. Operating Records. — The operating records of any plant bear the same relationship to the economical operation of that plant as the bookkeeping and cost accounting system bears to the manufacturing plant. The distribution of profit and loss in either case can only be obtained by itemizing the various factors involved and by grouping them in such a manner as to show at any time where improvement is possible. Commercial bookkeeping has been more or less standardized

and entails very little need of originality on the part of the bookkeeper, but the selection and maintenance of a system of power-plant records may require considerable study and experimenting, since each installation is a problem in itself. The items included in the different forms depend upon the apparatus provided for weighing the coal and water,

TABLE 130.

PERMANENT STATISTICS.

MAIN ENGINES.

Make	Ball Engine Co.	Shop number	..
Tandem or cross compound	Cross Compound	Height over all, ft.	13
Number in plant	5	Width, ft.	14
Rated horse-power	250	Length, ft.	9
Average load	220	Dimensions of foundation, ft.	13.5×10.5
Minimum load	100	Material of foundation	Concrete
Maximum load	315		
Best economy, lbs. per h.-p. hr	22	Weight of engine, lbs.	50,000
Average economy, lbs. per h.-p. hr	25	Cost of engines and generators	$45,000
Steam pressure, gauge	150	Stroke, in.	16
Receiver pressure, gauge	40	Revolutions per min.	225
Exhaust pressure	Atmospheric	Weight of heaviest part, lbs.	28,000
Piston speed, ft. per min.	604	R.p.m. of governor	225
Type of governor	Robb-Armstrong-Sweet	Diam. of flywheel, in.	72
		Face of flywheel, in.	16.75
Speed variation, per cent.	1	Weight of flywheel, lbs.	4,000

	H.P.	L.P.
Diameter of cylinder, in.	14	22
Clearance, per cent	4.5	4.5
Steam pipe diameter, in.	5	7
Exhaust pipe diameter, in.	10	10
Area of the port opening, in.	10×1.75	14×2.5
Diameter and length of main bearing, in.	7×12	6×12
Diameter and length of crosshead pin, in.	4, 3	4, 3
Diameter and length of crank pin, in.	7 in., 3	7 in., 3
Diameter and length of main shaft	7 in., 14.5 ft.	7 in., 14.5 ft.
Diameter of piston rod	2.5 in.	2.5 in.
Kind of piston packing	Snap ring	Snap ring
Size of piston packing	$\frac{5}{8}$ in. square	$\frac{5}{8}$ in square
Kind of rod packing	Metallic	Metallic
Area crosshead surface, sq. in.	144.5	144.5

RECEIVER.

	No. Heating Coils.
Diameter, in.	7
Length, ft.	3.5
Volume, cu. ft.	1.05
Diameter of drain pipe	

the type and number of instruments available for measuring temperature, pressure, and power, and the system adopted for keeping track of oil, waste, general supplies, and repairs. In large stations autographic recording and integrating appliances, which are to be found in nearly

TABLE 131.

PERMANENT STATISTICS.

FEED PUMPS.

Date of installation		Diameter of steam cylinder	16
Make	Snow	Diameter of water cylinder	10
Number in plant	2	Stroke	12
Height, ft	3	Displacement per stroke, cu. ft	0.5454
Length, ft	12	No. of strokes per min., average	12
Width, ft	4	Diameter of suction	8
Weight of pump	5 tons	Diameter of discharge	5
Cost, each	$965	Diameter of steam pipe	2.5
Steam pressure	150	Diameter of exhaust	4
Back pressure	½	Diameter of steam drips	½
Number of valves	32	Diameter of water drains	½
Character of valves	Rubber, brass lined	Suction head, lb. per sq. in.	1½
Area thro' valve seats, sq. in., per pump	12.13	Discharge head, lb. per sq. in.	175
Gallons of water per min., per pump	800	Kind of piston packing	Outside packed plunger
Pounds of water per 24 hrs., average, actual	479,400	Size of piston packing	....
Gallons of water per 24 hrs	599.2	Kind of rod packing	Soft
Volume of air chamber, cu. ft.	3	Size of rod packing	⅝
Shop number	24,572–3	Temperature of feed water	214

all strictly modern stations and represent but a small part of the first cost of the plant, greatly reduce the labor of keeping continuous records. In small plants the cost of autographic instruments may prove to be prohibitive and recourse must be had to the usual indicating devices. In the latter case, continuous records may be closely simulated by plotting the readings of the indicating appliances, say every 15 minutes, or even once every hour, and by connecting the points with a straight line. (See Figs. 581 to 583.) The oftener the readings are taken the smaller will be the error. Total quantities may be obtained by summing up the various items or by integrating the graphical chart by means of a planimeter. It is not sufficient to record monthly or yearly averages. Daily and even hourly records are absolutely essential for maximum economy. The various losses may be reduced to a minimum only by an intelligent analysis of daily records. A number of forms taken from the files of various power plants are reproduced in this chapter under the proper subheadings and serve to illustrate current practice.

Power Plant Records: Prac. Engr. U. S., July 1, 1912, p. 668; March 1, 1912, p. 242; Jan. 1, 1912, p. 36. Power, May 28, 1912, p. 758.

462. Output and Load Factor. — The output of a plant is usually stated in terms of the (1) average horse power, or equivalent, for a given period of time. (2) Unit output — horse-power hours, or equivalent.

When the plant is operating at practically constant load it is sufficiently accurate for most purposes to express the output in horse power, or equivalent, per month or per year. When the output fluctuates as is the general case, it is best expressed in terms of unit output. For example, one horse power per year, 24 hours per day, and 365 days per year is equivalent to $365 \times 24 = 8760$ horse-power hours. If the full power is used throughout this time it matters little whether the charge is based on the *flat rate* (horse power per year) or the *unit rate* (horse-power hours); if, however, the power is used only half the time, the yearly cost per horse-power hour will be just double.

The *yearly load factor* or simply *load factor* is the ratio of the *actual* yearly output to the *rated* yearly output measured on the twenty-four-hour basis. Thus:

$$\text{Load factor} = \frac{\text{Yearly output, horse-power hours or equivalent}}{\text{Rated horse power, or equivalent} \times 8760}. \qquad (291)$$

The *curve load factor* or *station load factor* is the ratio of the yearly output to the rated output based upon the number of hours the plant is in actual operation. Thus, for an electric station:

$$\text{Curve load factor} = \frac{\text{Yearly output, kilowatt-hours}}{\text{Rated capacity} \times \text{hours plant is in operation}}. \qquad (292)$$

Much confusion arises from the interpretation of the term "rated capacity." If rated below the maximum load it can sustain it is evident that a prime mover may operate with a load factor over 100 per cent, in which case the term is without purpose. The accepted definition of rated load in this connection is the maximum load which the prime mover can sustain continuously on a twenty-four-hour basis without overheating. Other definitions have been assigned to the term load factor and station factor, but the two stated above are more in accord with current practice.

In any plant the great desideratum is a high load factor with greatest return on the investment. All the factors of expense included in the cost of power are then operating at maximum economy. High peak loads and low average loads necessitate large machines which are but little used and greatly increase the fixed charges.

The *demand factor* is the ratio of the maximum demand to the connected load. There is a general tendency to overestimate the maximum electric demand, due, in a measure, to the possibilities of all the

lights and motors being in use at one time. Practically speaking, such conditions are not likely to occur. Table 132 gives an idea of the value of the demand factor for various classes of service and may be used as a guide for determining the size of prime movers.

TABLE 132.

CENTRAL STATIONS, DEMAND FACTORS.*

Demand factors compiled by Commonwealth Edison Company of Chicago.

CLASS OF SERVICE.

	Demand Factor.
Lighting customers:	
Billboards, monuments, and department stores	85.6
Offices	72.4
Residences and barns	60.0
Retail stores	66.3
Wholesale stores	70.1
Average	59.8
Motor customers:	
Offices	65.1
Public gathering places and hotels	28.7
Residences and barns	69.3
Retail stores	61.2
Wholesale stores and shops	58.2
Average	59.4

Demand factors compiled by Wisconsin State Commission from companies using Wright Demand Meter.

CLASS OF SERVICE.

	Demand Factor. Min. Max.
Churches	56- 85
Clubs	28
County and Federal buildings	31- 33
Depots	75- 95
Factories	53- 56
Hotels	28
Laundries	60- 75
Livery stables	52- 58
Lodge and dance halls	68
Offices	57- 87
Restaurants	52- 62
Saloons	62- 92
Schools	37- 52
Shops	55
Shops — blacksmith	66
Shops — machine	37- 54
Stores	40-100
Theatres	49- 89

TABLE 133.

TYPICAL OPERATING CHART

DAILY

COAL TICKET.

No......... ....191.....

6 a.m. to 6 p.m.		6 p.m. to 6 a.m.	
Lbs. Coal.	Lbs. Ashes.	Lbs. Coal.	Lbs. Ashes.
....	....	....	....
....	....	....	....
....	....	....	....
....	....	....	....
....	....	....	....
....	....	....	....
....	....	....	....
....	....	....	....
....	....	....	....
....	....	....	....
....	....	....	....
Kind of Coal.		Kind of Coal.	
Fireman.		*Fireman.*	

Coal used ....lbs.
Ashes made...............lbs.
Per cent ashes.............lbs.
Water used.................gals.
Water used.................lbs.
Water.........lbs. to 1 lb. Coal.
Boiler Water Test: Good. Med. Low.

TABLE 134.

TYPICAL OPERATING CHART.

DAILY POWER-HOUSE REPORT.

THE UNITED LIGHT AND POWER CO.

..................Division

....19..

Weather — Noon

			Hr.	Min.
Engine No. 1	started...........M	stopped..........M	Total time run.........	......
Engine No. 2	started...........M	stopped..........M	Total time run..........	......
Inc. current on	..................M	off ....M	Total time on..........	......
Street arcs on	..................M	off...............M	Total time on..........	......

Noon AMPERE READINGS.

12 00	12 30	1 00	1 30	2 00	2 30	3 00	3 30	4 00	4 15	4 30	4 45	5 00	5 15	5 30	5 45	6 00	6 15	6 30
6 45	7 00	7 15	7 30	7 45	8 00	8 15	8 30	8 45	9 00	9 15	9 30	9 45	10 00	10 30	11 00	11 30	12 00	1 00
2 00	3 00	4 00	5 00	5 15	5 30	5 45	6 00	6 15	6 30	6 45	7 00	7 15	7 30	7 45	8 00	9 00	10 00	11 00

	Coal Received on Track.	Boilers in Service.
Coal used.....................lbs.	Car No.........................	No. 1 from........m to.......m
Cylinder oil ....pts.	Initial.........................	No. 2 from........m to.......m
Engine oil ....pts.	Time placed ....m	No. 3 from........m to.......m
Waste.........................lbs.	Time released................m	Washed No......................
Water.....................cu. ft.	Weight.....................lbs.	Blew No........................
Carbons.........................	Ashes sold ....loads to	....
Globes..........outer....inner..		

Material Received for Power House Use.	Total Kilowatt Output. Read meter 12 o'clock noon
....	
....	Meter to-day............... Kw.
....	Meter yesterday............ Kw.
....	Diff...........................

Report here ANY interruption of service either arc or incandescent.

Time off..............Cause...

Arc lights out ....

Lights..

Location Reported by

TABLE 135.

TYPICAL OPERATING CHART.

(Large Chicago Department Store.)

Monthly Report. ....19..

Date.	Average Outside Temperature.	Fuel.					Supplies.			
		Coal.				Ash.	Oil Used, Gals.		Waste Pounds.	Total Water to Building, Cu. Ft.
		Kind.	Pounds Burned.	Cost Per Ton.	Cost Per Day.	Pounds Removed.	Engine.	Cylinder.		

Output.				Engine-Hours Run.					Boilers-Hours Run.						Breeching.	
Boilers.		Generators.														
Pounds of Water Evaporated.	Water Evaporated Per Lb. of Coal.	Ampere-Hours.	Kilowatt-Hours.	1	2	3	4	5	1	2	3	4	5	6	Draft.	Temperature.

Heating System.		Ventilating Plants, Hours Run.		Refrigerating Plant.			Repairs-Hours.		
Steam Pressure.	Live Steam-Hours.	Fan 1	Fan 2	Hours Run.	Gas Used, Pounds.	Ice Made, Pounds.	Engine Room.	Boiler Room.	Miscellaneous.

In the original copy all of these items are conveniently grouped on one large form ruled for 31-day entries with space at bottom for total quantities and costs. In the reproduction only the headings are included.

TABLE 136.

TYPICAL OPERATING CHART.

ANNUAL STATEMENT, FIRST NATIONAL BANK BUILDING, CHICAGO. (YEAR 1911.)

Expenditures.

Coal	$39,798.64	Supplies:	
Ash cartage	2,669.25	Oil waste and grease	$1,728.96
Water	1,296.00	Packing	500.00
Supplies and repairs:		Plumbing	463.79
Elevators	1,948.08	Lamps	2,377.16
Engine room	631.80	Wages	26,986.84
Boiler room	2,121.12	Petty cash	27.83
Supplies:		Office	119.75
Electrical	1,952.01	Coal analysis	240.00
Refrigerating	1,129.22	Machine shop	183.42
Steam fitting	296.63		
Heating	2,455.02	Total	$87,891.24

TABLE 136 (*Continued*).

Credits.

Receipts from sale of current, steam power and the like..	$75,636.77
Net cost of operation..............................	1,225.47
Assumed credits, account of elevator service, steam heating, etc., for which payment is not made..............	70,260.39
Net earnings including all credits........................	58,005.92

Unit costs — all expenses charged to power.

			Cents.
Pounds of coal, total for year.....	31,459,220	Labor per kw.-hr.....	1.45
Total output, kilowatt-hours.....	1,854,400	Coal per kw.-hr......	2.14
Coal per kilowatt-hour...........	16.9	Supplies per kw.-hr...	1.13
		Total...........	4.74

463. Cost of Operation. — The cost of operation of power plants is conveniently divided into two parts:

(1) Fixed charges.
  (*a*) Investment costs.
  (*b*) Administration costs.

(2) Operating costs.

464. Fixed Charges. — These cover all expenses which do not expand and contract with the output. In very large plants they are usually divided into two parts: (*a*) the *investment costs*, which include interest, rental, depreciation, taxes, and insurance, and a reserve fund to cover depreciation of the investment, and (*b*) the *administration costs*, which include rental of offices, annual salaries of officers, and all other expenses not directly chargeable to the power plant. In the average plant the fixed charges comprise interest, rental, depreciation, taxes, insurance, and sometimes maintenance, though the latter is ordinarily included in the operating costs.

In any system the total fixed charges per year are constant irrespective of the load factor, since interest, taxes, depreciation, insurance, and maintenance go on whether the plant is in operation or not. The total fixed charges for a specific case are illustrated in Fig. 581 by a straight line. The cost per kilowatt-hour, however, decreases as the load factor increases. For example, with the plant operating continuously at rated load (100 per cent load factor) the fixed charges per kilowatt-hour are

$$\frac{65,000}{5000 \times 8760} = \$0.00148.$$

With 30 per cent load factor these charges are

$$\frac{65{,}000}{0.3\ (5000 \times 8760)} = \$0.00445 \text{ kilowatt-hour.}$$

The higher the load factor the greater is the amount of power produced and the longer does the apparatus work at best efficiency. But the greater the power produced the larger will be the fuel consumption and the oil and supply requirements. The labor charges will be practically constant. The total operating cost per year increases as the load factor increases, but not directly. (See Fig. 581.) The cost per kilowatt-hour, however, decreases as the load factor increases. For

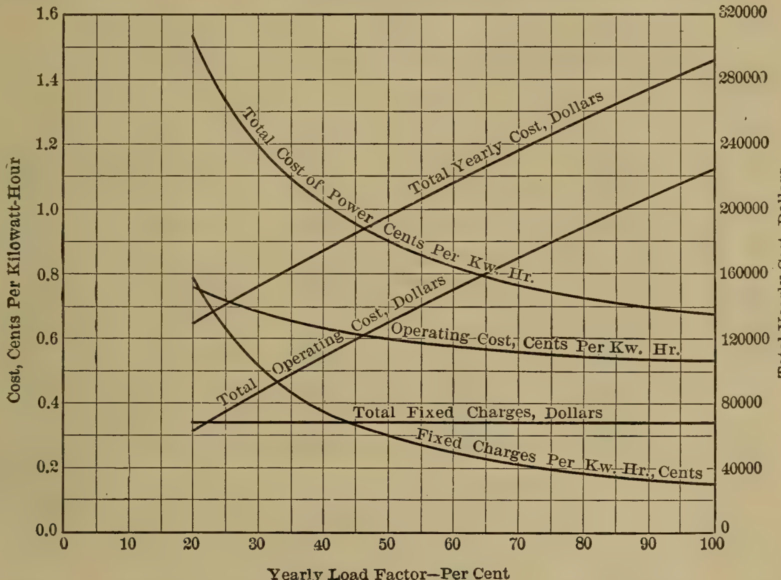


FIG. 581. Influence of Load Factor on the Cost of Power at the Switchboard. (5000-kilowatt Electric Light and Power Station.)

example, the operating costs per year with plant operating continuously at full load are \$230,200. This gives

$$\frac{230{,}200}{5000 \times 8760} = \$0.00525 \text{ per kilowatt-hour.}$$

With 30 per cent load factor the yearly operating charges are \$87,980, which gives

$$\frac{87{,}980}{0.3\ (5000 \times 8760)} = \$0.0067 \text{ per kilowatt-hour.}$$

Table 149 shows the influence of the load factor on the cost of power in two isolated stations of the same rated capacity, one operating

with the unusually high load factor of 80 per cent and the other operating with the low load factor of 17 per cent. The former furnishes current for a large electro chemical concern in which the load is practically constant.

In general, the higher the load factor the greater becomes the ratio of the operating to the fixed charges, and extra investment may become advisable to secure the greatest economy possible.

TABLE 137.

AVERAGE INITIAL COST.*

Steam Engine Power Plants.

Simple Non-Condensing.

Horse Power.	Dollars per Horse Power.	Horse Power.	Dollars per Horse Power.
10	225.00	60	180.00
20	200.00	70	177.00
30	195.00	80	175.00
40	190.00	90	170.00
50	185.00	100	165.00
Compound Condensing.			
100	170.00	700	76.00
200	146.00	800	69.00
300	126.00	900	64.00
400	110.00	1000	60.00
500	96.00	1500	58.00
600	85.00	2000	55.00
Triple Condensing.			
1000	62.00	4000	52.00
2000	58.00	5000	50.00
3000	54.00	6000	48.00

* Includes cost of buildings and entire equipment erected.

On the other hand, when the load factor is low the fixed charges are the governing factor in the cost of power, and extra expenditures must be carefully considered, particularly if fuel is cheap.

Fixed Costs in Industrial Power Plants: Engineering Digest, Apr., 1911, p. 293.

465. Interest. — The rates of interest on borrowed money vary with the nature of the security. In the case of power plants the form of security is usually a mortgage on the plant and equipment. If a builder has sufficient funds to construct the plant without borrowing, he should charge against the item "interest" the income which the sum

involved would bring if placed out at interest or if invested in his own business. In estimating the interest charges 5 per cent of the capital invested is ordinarily assumed unless specific figures are available.

466. Depreciation. — This charge represents the gradual deterioration of a plant, resulting in its eventually wearing out. It is also assumed to represent the superannuation of a plant or the rate at which the apparatus is becoming obsolete. Thus, under the first assumption, if the useful life of an engine is 40 years, the rate of depreciation, neglecting interest, is 2.5 per cent; if, however, it is assumed

TABLE 138.

COST OF MECHANICAL EQUIPMENT — ISOLATED STATIONS.*

	Per Kilowatt of Plant Capacity.
Boilers (erected and set in masonry):	
Horizontal-tubular	$14–$18
Water-tube	16– 20
Steam engines:	
High-speed, simple direct-connected	20– 25
Medium-speed, compound non-condensing direct-connected	28– 35
Low-speed, compound condensing, belted	20– 25
Low-speed, simple, belted	25– 30
Gas engines	50– 60
Oil engines	75– 85
Gas producers	15– 20
Dynamos:	
Direct-connected to high-speed engine	13– 16
Belt-connected to engine	12– 15
Direct-connected to Corliss engine	16– 20
Switchboard	5– 10
Foundations	5– 10
Steamfitting—including auxiliary apparatus—such as feed heater, grease separator, exhaust head, tanks, covering, etc	20– 30

* By P. R. Moses before the A.I.E.E., Jan. 12, 1912.

that the engine will become obsolete in 20 years and uneconomical for further operation, the rate of depreciation will be 5 per cent. It is difficult to assign a fixed rate of depreciation against any piece of apparatus, due to possible new developments which cannot be reckoned with in advance in computing the useful life of the apparatus. Again, depreciation cannot always be separated from current repairs and is a variable factor even in the parts of the same machine. It is, therefore, more or less of an approximation. The average life of various parts of a steam power plant is outlined in Table 140, but on account of the inability to assign fixed values to the useful life of any apparatus, and on account of the great number of appliances in even a small plant, it is customary to charge a fixed rate of depreciation against the entire

plant and thus avoid confusion and complexity. This very crude method usually results in overestimation in well-designed, well-operated plants and underestimation in poorly designed and badly managed installations.

TABLE 139.

COST OF MECHANICAL EQUIPMENT — STEAM TURBO-ELECTRIC GENERATING STATIONS.*

2,000 to 20,000-kilowatt Capacity, Based on Maximum Continuous Capacity of Generators at 50° Rise.

	Dollars per Kilowatt.	
	High.	Low.
Preparing site — Dismantling and removing structures from site, making construction roads, tracks, etc.	$0.25	$....
Yard Work — Intake and discharge flumes for condensing water, railway siding, grading, fencing sidewalks	2.50	1.00
Foundations — Including foundations for building, stacks, and machinery, together with excavation, piling, waterproofing, etc.	6.00	1.00
Building — Including frame, walls, floors, roofs, windows and doors, coal bunker, etc., but exclusive of foundations, heating, plumbing, and lighting	12.00	4.00
Boiler-room Equipment — Including boilers, stokers, flues, stacks, feed pumps, feed-water heater, economizers, mechanical draft, and all piping and pipe covering for entire station except condenser water piping	24.00	12.00
Turbine-room Equipment — Including steam turbines and generators, condensers with condenser auxiliaries and water piping, oiling system, etc.	22.00	12.00
Electrical Switching Equipment — Including exciters of all kinds, masonry switch structure with all switchboards, switches, instruments, etc., and all wiring except for building lighting	5.00	2.00
Service Equipment — Such as cranes, lighting, heating, plumbing, fire protection, compressed air, furniture, permanent tools, coal- and ash-handling machinery, etc.	5.00	2.50
Starting Up — Labor, fuel, and supplies for getting plant ready to carry useful load	1.00	.50
General Charges — Such as engineering, purchasing, supervision, clerical work, construction, plant and supplies, watchmen, cleaning up	6.00	3.00
Total cost of plant to owner, except land and interest during construction	$83.75	$38.00

* By O. S. Lyford, J ., and R. W. Stoval, of Westinghouse, Church, Kerr & Company, before the Engineer's Society of Western Pennsylvania.

In some of the very large stations depreciation is divided and charged as follows:

(*a*) *Complete depreciation* due to wear and tear. This is charged to the proper maintenance account.

(*b*) *Obsolescence, inadequacy* or *destruction* by any cause. This is written off from current income or from past accumulations of profit and loss account, or it is capitalized in the usual way.

(*c*) *Incomplete depreciation,* due to wear and tear, likely to fall in large amounts at irregular intervals. This is provided for by a depreciation fund.

The depreciation fund is essentially a financial problem and the proper amount to be charged varies with every property, and should be carefully estimated either as a factor of the physical cost or of the gross earnings.

TABLE 140.

APPROXIMATE USEFUL LIFE OF VARIOUS PORTIONS OF STEAM POWER PLANT EQUIPMENTS.

	Years.
Buildings, brick or concrete	50
Buildings, wooden or sheet iron	15
Chimneys, brick	50
Chimneys, self-sustaining steel	25
Chimneys, guyed sheet-iron	10
Boilers, water-tube	25
Boilers, fire-tube	15
Engines, slow-speed	25
Engines, high-speed	15
Turbines	25
Generators, direct-current	25
Generators, alternating-current	30
Motors	20
Pumps	25
Condensers, jet	35
Condensers, surface	20
Heaters, open	30
Heaters, closed	20
Economizers	20
Wiring	20
Belts	7
Coal conveyor, bucket	15
Coal conveyor, belt	10
Transformers, stationary	30
Rotary converters	25
Storage batteries	15
Piping, ordinary	12
Piping, first class	20

NOTE.— So much depends upon the design and the conditions of operation that no fixed values can be definitely assigned and the above figures should be used with caution. Practice shows that most power-plant appliances become obsolete long before the limit of their useful life is reached.

The rate of depreciation in terms of interest and useful life is a simple problem in compound interest, and may be expressed:

$$d = \frac{100\,r}{(1 + r)^n - 1}, \tag{293}$$

in which

d = rate of depreciation, per cent of first cost.

r = rate of interest.

n = assumed life of the apparatus in years.

This is based on the assumption that the interest is compounded annually and that the apparatus is valueless at the end of n years. Table 141 has been calculated with this formula and gives the rate of depreciation for different rates of interest and different assumptions as to useful life of apparatus.

TABLE 141.

RATE OF DEPRECIATION.

(Per Cent of First Cost.)

Assumed Useful Life of Apparatus.	Rate of Interest, per Cent.												
	2	2.5	3	3.5	4	4.5	5	5.5	6	7	8	9	10
2	49.50	49.37	49.27	49.14	49.02	48.90	48.78	48.66	48.54	48.31	48.07	47.84	47.62
3	32.67	32.51	32.35	32.19	32.03	31.87	31.72	31.56	31.41	31.10	30.80	30.51	30.21
4	24.26	24.08	23.90	23.72	23.55	23.39	23.20	23.03	22.86	22.52	22.19	21.84	21.55
5	19.21	19.02	18.83	18.65	18.46	18.28	18.10	17.91	17.73	17.40	17.04	16.73	16.37
6	15.85	15.65	15.46	15.26	15.08	14.89	14.70	14.52	14.33	13.97	13.63	13.29	12.96
7	13.45	13.25	13.05	12.85	12.66	12.46	12.28	12.09	11.91	11.15	11.20	10.87	10.55
8	11.65	11.44	11.24	11.05	10.85	10.66	10.47	10.28	10.10	9.74	9.40	9.06	8.74
9	10.25	10.04	9.84	9.64	9.45	9.26	9.07	8.88	8.70	8.34	8.00	7.68	7.36
10	9.13	8.92	8.72	8.52	8.33	8.14	7.95	7.76	7.58	7.23	6.90	6.58	6.27
11	8.21	8.01	7.80	7.61	7.41	7.22	7.04	6.85	6.68	6.33	6.00	5.69	5.40
12	7.45	7.25	7.04	6.85	6.65	6.46	6.28	6.10	5.92	5.60	5.27	4.97	4.69
13	6.81	6.60	6.40	6.20	6.01	5.83	5.64	5.47	5.29	4.96	4.65	4.36	4.08
14	6.26	6.05	5.85	5.65	5.46	5.28	5.10	4.93	4.75	4.49	4.13	3.84	3.58
15	5.78	5.57	5.37	5.18	4.99	4.81	4.63	4.46	4.29	3.97	3.68	3.40	3.15
16	5.36	5.16	4.96	4.77	4.58	4.40	4.22	4.06	3.89	3.58	3.30	3.03	2.78
17	4.99	4.79	4.59	4.40	4.22	4.04	3.87	3.70	3.54	3.24	2.96	2.71	2.47
18	4.67	4.46	4.27	4.08	3.90	3.72	3.55	3.39	3.23	2.94	2.66	2.42	2.19
19	4.37	4.17	3.98	3.79	3.61	3.44	3.27	3.11	2.96	2.67	2.47	2.17	1.95
20	4.11	3.91	3.72	3.53	3.36	3.19	3.02	2.87	2.71	2.44	2.18	1.95	1.95
25	3.12	2.92	2.74	2.56	2.40	2.24	2.09	1.95	1.82	1.58	1.36	1.18	1.75
30	2.46	2.27	2.10	1.93	1.78	1.64	1.50	1.38	1.26	1.06	0.88	0.73	0.61
35	2.00	1.82	1.65	1.50	1.36	1.23	1.10	0.99	0.89	0.72	0.58	0.46	0.37
40	1.65	1.48	1.32	1.18	1.05	0.93	0.83	0.73	0.64	0.50	0.38	0.29	0.22
45	1.39	1.22	1.07	0.94	0.82	0.72	0.62	0.54	0.47	0.35	0.26	0.19	0.14
50	1.18	1.02	0.88	0.76	0.65	0.56	0.42	0.40	0.34	0.25	0.17	0.12	0.09

Example: A 1000-square-feet surface condenser and auxiliaries cost \$3500. With interest at 5 per cent, required the rate of depreciation, assuming a life of 25 years.

$$d = 100 \frac{0.05}{(1 + 0.05)^{25} - 1} = 2.09 \text{ per cent.}$$

That is to say, if 2.09 per cent of the first cost is laid aside each year for 25 years at 5 per cent interest, compounded annually, it will equal the first cost at the end of this period. The sum thus laid aside is sometimes called the *sinking fund.* The solution is more readily obtained with the aid of Table 141; e.g., at the intersection of vertical column 5 (interest) and horizontal column 25 (life in years) we find the depreciation 2.09 per cent. This sinking-fund method of rating

TABLE 142.

DEPRECIATION PERCENTAGES DETERMINED BY VARIOUS AUTHORITIES.*

Items.	Chicago Traction Commission.	Chicago Union Traction Co.	Milwaukee Electric Railway & Light Co.	Wisconsin Railroad Commission.	Average English.	Average Scotch Practice.	Philip Dawson.	Stone and Webster.	Industrial Power Plants.	G. F. Gebhardt.	Miscellaneous Sources.	Average for 11 authorities given in preceding columns.
Buildings	......	2	3	2	2.5	2.5	1–2	2	1–2	2	2	1.9
Boilers	3.5–10	6.6	7.5	6.6–8.5	5	5	8–10	5	2.5–3.3	4–6.6	7.5	6.2
Steam piping	3.5	6.6	7.5	5	5	5	8–10	5	2.5–3.3	5–8	5	5.54
Auxiliaries	5–10	6.6	5	6.6–8.5	5	5	8–10	5	4–6.6	3–5	7.5	6.13
Steam engines	3–10	6.6	5	5–6.6	5	5	4–6	5	2.5–5	4–6.6	5	5.26
Steam turbines	......	......	5	....	5	5	7–9	5	2.5–5	4	4	4.96
Belted generators	5–10	6.6	7.5	5	5	5	5–10	5	6.6	3.3–4	7.5	6.07
Direct-connected generators	5	6.6	7.5	5	5	5	4–8	5	4	3.3–4	5	5.25
Wires and cables	2	6.6	5	2	5	3	3–5	5	4–6.6	5	5	4.36
Switchboards, etc	2	6.6	5	2	7.5	5	8–10	5	2–5	......	5	5.06
Rotary converters	......	6.6	5	5	5	5	8–10	5	4–5	4	5	5.41
Transformers	......	6.6	5	5	5	5	5–6	5	3–5	4	5	5.01
Motors	5–10	6.6	5	5	5	5	5–8	5	4–6.6	5	5	4.63
Storage batteries	......	6.6	10	....	5	5	9–11	5	5–10	6.6	10	7.3
Overhead systems	......	10–14	7.5	....	......	3	4–8	10–14	5–10	......	10	8.3
Cars	......	5–8.5	6–7.5	5–6.6	10	7.5	4–6	5–8.5	....	......	7.5	7.
Track work	......	7.57	7.5–12	5	5	8	7–13	7.2	....	......	7.5	7.53
Shop equipment	3–10	5	7.5	3.3–10	7.5	7.5	12–15	5	4–10	......	7.5	7.37
Supplies and miscellaneous	5	......	5	....	5	7.5	1.5–2	......	5	......	5	4.89

* From Elec. Review & Wes. Elecn.

the depreciation is peculiarly adapted to power-plant practice, inasmuch as a sum set aside at comparatively low rates of interest and compounded increases very slowly at first but more and more rapidly from year to year. This is precisely what happens in the deterioration of the plant. The loss of value is slight at first when the materials are new and usefulness is at a maximum, while towards the end of life both value and usefulness decline very rapidly.

If the apparatus still has some value at the end of n years and if b is the ratio of the value of the old material to that of the new, the rate of depreciation becomes

$$d = 100 \frac{r(1 - b)}{(1 + r)^n - 1}. \tag{294}$$

Example: In the foregoing problem, required the rate of depreciation if the value of the condenser outfit is \$350 at the end of 25 years.

$$\text{Here } b = \frac{350}{3500} = 0.1.$$

$$d = \frac{100 \times 0.05 (1 - 0.1)}{(1 + .05)^{25} - 1}$$

$$= 1.97 \text{ per cent.}$$

That is, 1.97 per cent of \$3500, or \$68.95, laid aside each year for 25 years at 5 per cent interest and compounded annually will equal \$3500 − 350, or \$3150, at the end of this period.

It is not supposed that an owner will regularly lay aside this sum annually, or take the trouble to arrange for its investment at current rates in the market or savings bank, since the money is probably worth more to him in his own business. In practice it is retained in his business or investments and is earning the rate of interest obtainable therein, but in determining the net profit or loss this depreciation item is nevertheless accounted for just as if it were actually placed in outside investments.

In appraising the present value of any apparatus in terms of the rate of interest and useful life the expression becomes

$$V = 100 \frac{(1 + r)^m - 1}{(1 + r)^n - 1}, \tag{295}$$

in which

V = total depreciation, per cent of original cost.
m = number of years apparatus has been in operation.
n = assumed life of apparatus.
r = rate of interest.

Example: In the preceding example, required the present valuation of the condenser, assuming that it has been in use 15 years.

$$m = 15, n = 25, r = 0.05.$$

Substituting these values in (295),

$$V = 100\frac{(1 + 0.05)^{15} - 1}{(1 + 0.05)^{25} - 1} = 45.1 \text{ per cent.}$$

That is, the condenser has depreciated 45.1 per cent of its original value and consequently is worth $\$3500 - 0.451 \times 3500 = \$1921.50$.

Table 141 may be conveniently used in this connection. At the intersection of vertical column 5 and horizontal columns 15 and 25 we find 463. and 2.09 respectively. Dividing 2.09 by 4.63 we get 0.451 = 45.1 per cent, the total depreciation. Depreciation is often taken care of under the different items pertaining to maintenance, and whenever a change or repair is necessary it is charged directly into expense as maintenance.

Though usually considered separately, interest and depreciation are sometimes considered as a single item, and in this case the rate of depreciation represents the sum which must be laid aside each year for the eventual renewal of apparatus plus the interest on the investment.

Depreciation: Jour. Elec. Power & Gas, Feb. 25, 1911, p. 176; Elec. Rev. & Wes. Elecn., Apr. 23, 1910, p. 852, Dec. 2, 1911, p. 1126; Elec. Wld., Dec. 2, 1909, p. 1349; Power, June 18, 1912, p. 878, Feb. 25, 1911, p. 176, Nov. 22, 1910, p. 2087, Aug. 9, 1910, p. 1439, July 5, 1910, p. 1227.

467. Maintenance. — Maintenance usually refers to the expense of keeping the plant in running order over and above the cost of attendance. It includes cost of upkeep, replacement, and precautionary measures. This latter item includes the renewal of working parts, painting of perishable or exposed material, and replacing worn-out and defective material. Many engineers make no allowance for maintenance in the fixed charges and include these costs under supplies, attendance, or repairs. In a general way, when maintenance is included under the fixed charges, an annual charge of 2 per cent is considered a liberal allowance, since most of the repair work comes under attendance. In street-railway practice maintenance is divided among the several parts of the system as follows: Buildings, steam appliances, electrical equipment, and miscellaneous. In this connection the maintenance becomes a part of the operating charges, since the various items vary widely from month to month.

468. Taxes and Insurance. — Taxes vary from a fraction of one per cent to one and one-half per cent, depending upon the location of the plant. An average figure is one per cent of the actual value of the

investment. Buildings and machinery are ordinarily insured against fire loss and boilers against accidental explosions, and accident policies are sometimes carried on all operating machinery. A fair charge for this item is one-half per cent.

469. Operating Costs. — Operating costs are conveniently divided as follows:

(1) Labor or attendance.
(2) Fuel and water.
(3) Oil, waste, and supplies.
(4) Repairs and maintenance.

In large stations it is often desirable to keep the expenses of the various departments separate from those of the power-plant proper. Thus in central stations the distributing system is an important branch and the attending expenses form a considerable portion of the total. They are therefore kept separate, since they are not strictly chargeable to power generation. In isolated stations the wages of elevator men, though in a general way a part of the power-plant expenses, are not included in the "labor and attendance" charge of the plant. Lamps are a large item of expense in a tall office building, and for this reason are often kept separate from supplies.

470. Labor, Attendance, Wages. — The minimum number of men required to handle a given plant is approximately a fixed quantity and it is seldom possible to so arrange the work that any material reduction can be effected. Until very recently it has been the universal custom to pay wages on a "flat rate" basis, that is, the attendant is given a fixed sum per day or month irrespective of the amount of work required or the economy of operation. In many cases, however, the bonus system has been successfully adopted. For example, in the boiler room the coal consumption is determined for a given period of time with ordinary careful firing, and the fireman is offered a reasonable percentage on the saving of coal which he is able to effect over this record by special care and attention to the keeping of fires always in the best condition, avoiding the blowing off of steam, using as little coal as needed for banking fires, and in other ways. Where careful records are kept of supplies, repairs, and renewals, the bonus is also applicable to electricians, oilers, and other employees.

Labor should always be estimated or recorded as so many dollars per month or per year and not merely in terms of the output unless the load factor is definitely known, otherwise comparisons are misleading. For example, consider two plants of 500 kilowatts capacity, each with labor charges, say, of $400 per month. Suppose the output

TABLE 143.

COST OF LABOR FOR STREET-RAILWAY PLANTS.

(C. C. Moore & Co.)

Size of Plant, Kw.	Chief Engineer. No.	Chief Engineer. Rate per Month	Assistant Engineer. No.	Assistant Engineer. Rate per Month	Watch Engineer. No.	Watch Engineer. Rate per Month	Boiler-room Engineer. No.	Boiler-room Engineer. Rate per Month	Oilers. No.	Oilers. Rate per Month	Firemen. No.	Firemen. Rate per Month	Boiler Cleaners. No.	Boiler Cleaners. Rate per Month	Wipers. No.	Wipers. Rate per Month	Water Tenders. No.	Water Tenders. Rate per Month	Electricians. No.	Electricians. Rate per Month	Switchboard Men. No.	Switchboard Men. Rate per Month	Generator Men. No.	Generator Men. Rate per Month	Clerks. No.	Clerks. Rate per Month	Total Cost of Labor per Year.	Cost of Labor per Kw. per Year.	Cost of Labor in Dollars per Kw. Hour. Yearly Plant Load Factor based on 365 Days, 24 Hours per Day=8760 hours per Year. .25	.33⅓	.50	.75	1.00
100	1	$85	...	...	1	$80	...	...	...	...	1	$60	...	...	...	...	...	...	...	...	...	...	...	...	...	...	$2,700	$27.00	.01233	.00925	.00616	.00410	.00308
200	1	85	...	...	1	80	...	...	...	...	1	60	...	...	...	...	...	...	...	...	...	...	...	...	...	...	2,700	13.50	.00616	.00462	.00308	.00205	.00154
250	1	85	...	...	1	80	...	...	...	...	1	60	...	...	...	...	...	...	...	...	...	...	...	...	...	...	2,700	10.80	.00493	.00370	.00247	.00164	.00123
300	1	90	...	...	1	80	...	...	...	...	1	60	...	...	...	...	...	...	...	...	...	...	...	...	...	...	2,760	9.20	.00420	.00315	.00210	.00140	.00105
400	1	90	...	...	1	80	...	...	...	...	1	60	...	...	1	60	...	...	...	...	...	...	...	...	...	...	3,480	8.70	.00397	.00298	.00199	.00132	.00099
500	1	90	...	...	1	85	...	...	1	60	1	60	...	...	1	60	...	...	...	...	...	...	...	...	...	...	4,260	8.52	.00389	.00292	.00195	.00130	.00097
750	1	90	...	...	1	85	...	...	1	60	2	70	...	...	1	60	...	...	...	...	...	...	...	...	...	...	5,220	6.96	.00318	.00238	.00159	.00106	.00079
1000	1	110	...	...	1	90	...	...	2	60	2	70	...	...	2	60	...	...	...	...	...	...	...	...	...	...	6,960	6.96	.00318	.00238	.00159	.00106	.00079
1500	1	120	...	...	1	90	...	...	2	60	2	70	1	65	2	60	...	...	...	...	...	...	2	60	...	...	9,300	6.20	.00283	.00212	.00142	.00094	.00071
2000	1	125	...	...	2	100	...	...	3	60	3	1–65 2–70	1	65	2	60	...	...	...	...	...	...	2	60	...	...	12,180	6.09	.00278	.00209	.00139	.00093	.00070
2500	1	125	...	...	2	100	...	...	4	60	3	1–65 2–70	1	70	2	60	...	...	...	...	...	...	2	70	...	...	13,200	5.28	.00241	.00181	00121	.00080	.00060
3000	1	125	...	...	2	100	...	...	4	65	3	70	3	2–60 1–70	2	60	...	...	...	...	...	...	2	70	...	...	14,940	4.98	.00227	.00171	.00114	.00076	.00057
4000	1	125	...	...	2	100	...	...	4	65	3	75	3	2–65 1 70	2	60	...	...	...	...	...	...	2	70	...	...	15,240	3.81	.00174	.00130	.00087	.00058	.00043
5000	1	135	...	...	3	100	...	...	5	65	3	75	4	3–65 1–70	3	60	...	...	...	...	1	75	3	1–65 2–70	...	...	20,520	4.10	.00187	.00141	.00094	.00062	.00047
7500	1	135	...	...	3	110	1	90	8	65	5	75	6	65	3	60	...	...	...	...	2	80	3	65	1	70	29,340	3.91	.00179	.00134	.00089	.00060	.00045
10000	1	150	1	120	2	120	1	90	10	65	6	75	7	65	4	60	3	75	...	...	3	1–85 2–80	2	65	2	1–65 1–70	37,560	3.76	.00172	.00129	.00086	.00057	.00043
15000	1	200	1	125	5	110	1	100	15	65	10	75	10	65	6	60	3	75	1	90	3	85	3	65	2	1–65 1–70	55,320	3.69	.00168	.00126	.00084	.00056	.00042
20000	1	250	2	125	5	110	1	100	20	65	12	75	12	2–70 10–65	8	60	6	75	1	90	3	85	5	65	3	2–65 1–70	71,280	3.56	.00162	.00122	.00081	.00054	.00041

NOTE. — Above figures approximate only, depending on size of units and conditions of service.

$$\text{Yearly load factor} = \frac{\text{Total kw. hrs. output per year}}{\text{Rated capacity kw.} \times 8760}.$$

All plants between 100 and 1500 kw. inclusive are direct current; all others alternating current.

All plants 4000 kw. capacity and smaller are operated continuously for 20 hours per day, 2–10 hour shifts; all others 24 hours per day, 3–10 hour shifts.

TABLE 144.

COST OF LABOR IN TALL OFFICE BUILDING POWER PLANTS.
SERVICE 18 HOURS, 2–10 HOUR SHIFTS.

(C. C. Moore & Co.)

Size of Plant, Kw.	Labor per Month when Heating Plant only is Installed.								Labor per Month when Complete Plant is Installed.																		Total Cost of Labor per Year with Plant.	Total Cost of Labor per Kw. Year with Plant.	Net Cost of Labor per Year Chargeable to Plant.	Net Cost of Labor per Kw. Year Chargeable to Plant.	Cost of Labor per Kw. Hour with Complete Plant, in Dollars. Load Factors.		Net Cost of Labor per Kw. Hour Chargeable to Plant, in Dollars. Load Factors.	
	Engineer.		Electricians.		Handy Man.		Pipe Fitter.		Chief Engineer.		Watch Engineer.		Oilers.		Firemen.		Boiler Cleaners.		Handy Man.		Wipers.		Switchboard Men.		Generator Men.						*.2 .1233	.4 .2466	*.2 .1233	.4 .2466
	No.	Rate	No.	Rate	No.	Rate	No.	Rate	No.	Rate	No	Rate	No.	Rate	No.	Rate	No.	Rate	No.	Rate	No.	Rate	No.	Rate	No.	Rate								
25	...	...	...	...	1	$50	...	...	1	$85	1	$75	...	...	...	...	...	...	...	...	...	...	...	...	...	...	$1,920	$76.80	$1,320	$52.80	.07111	.03556	.04889	.02444
50	...	...	...	...	1	50	...	...	1	85	1	75	...	...	...	...	...	...	...	...	...	...	...	...	...	...	1,920	38.40	1,320	26.40	.03556	.01778	.02444	.01222
75	...	...	...	...	1	50	1	50	1	85	1	75	...	...	...	...	...	...	...	...	...	...	...	...	...	...	1,920	25.60	720	9.60	.02370	.01185	.00889	.00444
100	1	75	...	...	1	50	...	...	1	90	1	85	...	...	...	...	...	...	...	...	...	...	...	...	...	...	2,100	21.00	600	6.00	.01944	.00972	.00556	.00278
150	1	75	...	...	1	50	...	...	1	90	1	85	...	...	1	70	...	...	...	...	...	...	...	...	...	...	2,940	19.60	1,440	9.60	.01815	.00907	.00889	.00444
200	1	85	...	...	1	60	...	...	1	100	1	85	...	...	1	70	...	...	...	...	...	...	...	...	...	...	3,060	15.30	1,320	6.60	.01417	.00708	.00611	.00306
250	1	85	...	...	1	60	1	60	1	100	1	85	...	...	2	70	...	...	...	...	...	...	...	...	...	...	3,900	15.60	1,440	5.76	.01444	.00722	.00533	.00267
300	1	85	...	...	1	60	1	60	1	100	1	85	...	...	2	70	...	...	...	...	...	...	...	...	...	...	3,900	13.00	1,440	4.80	.01200	.00600	.00444	.00222
400	1	100	1	60	1	60	1	60	1	100	1	85	...	...	2	70	...	...	1	65	1	65	...	...	...	...	5,460	13.65	2,100	5.25	.01264	.00632	.00486	.00243
500	1	100	1	60	1	60	1	60	1	100	1	90	1	60	2	75	1	60	1	65	1	60	1	75	...	...	7,920	15.84	4,560	9.12	.01467	.00733	.00844	.00422
750	1	100	1	60	2	60	1	60	1	100	2	85	1	60	2	75	1	60	1	70	1	60	1	75	...	...	8,940	11.92	4,860	6.48	.01104	.00552	.00600	.00300
1000	1	100	1	60	2	60	1	60	1	120	2	90	2	60	2	75	1	60	1	75	1	60	1	75	1	65	10,860	10.86	6,780	6.78	.01006	.00503	.00628	.00314

* Referring to above load factors, lower figures, .1233 and .2466, represent yearly load factors and are equal to $\frac{\text{Total kw. hrs. output per year}}{\text{Rated capacity kw.} \times 8760}$.

Upper figures, .2 and .4, represent daily curve load factors and are equal to $\frac{\text{Total kw. hrs. output per year}}{\text{Rated capacity} \times 5400}$ (based on 300 days per year, 18 hours per day).

Above figures approximate only, depending upon size of units and conditions of operation. All plants assumed to have boilers for heating purposes. When complete plants are installed for lighting and elevator service, all elevators assumed to be electric.

of one is 100,000 kilowatt-hours per month and that of the other 40,000 kilowatt-hours per month. The monthly charges are evidently the same, viz., $400, but the cost per kilowatt-hour differs widely, being 0.4 cent in the first case and 1 cent in the latter.

The cost of labor varies so much with the location of the plant and the conditions of operation that general figures are of little value except as a rough guide. The figures in Tables 143 and 144 give average results for general practice. Specific figures will be found in Tables 145 to 160.

For a summary of labor costs in large central stations see "Central-Station Labor Costs," Electrical World, Nov. 16, 1912, p. 1031.

471. Cost of Fuel. — Tables 145 to 160 give examples of the cost of fuel in different types and sizes of steam power plants. It will be noted that this item varies considerably even with plants of the same general class. So much depends upon the market price of the fuel itself that the item "cost per horse power or kilowatt-hour" gives little information concerning the economy of operation unless the price of the fuel, its calorific value, and the water rate of the prime movers are specified. In a general sense the cost of fuel will range from 40 to 70 per cent of the total operating expenses. In estimating the cost of fuel for a proposed installation due consideration should be given to the coal burned in banking fires, heat lost in blowing off boilers, and reduced efficiency in operating at underloads and overloads. For example, individual tests of a number of boilers in a large central station in Chicago gave an average evaporation of 6.1 pounds of water per pound of coal, actual conditions, whereas the evaporation determined by dividing the total water fed into the boiler per year by the total consumption of coal gave only 5.2 pounds. The difference between the two depends largely upon the standby losses in banking fires and in shutting down and starting up boilers. The following data from a number of installations in Chicago give an idea of the extent of these losses when burning Illinois coal:

1. Five 400-horse-power boilers with Murphy furnaces required 1300 pounds of coal for banking over night and 3800 pounds for firing up when cold.

2. Five 500-horse-power boilers with chain grates required 3900 pounds for banking 6 hours and 5000 pounds for banking over night and being put in service.

3. One 400-horse-power boiler with Dutch oven required 1500 pounds for 7-hour bank and 2500 pounds for 13-hour bank.

4. One 66 by 18 return tubular boiler, hand fired, required 500 pounds for 12-hour bank.

Current practice gives an average efficiency (based on yearly operation) of boiler and furnace of 70 per cent for pumping stations running at practically full load, 65 per cent for large lighting and power stations with yearly load factor of 0.50 or more, and 60 per cent for similar stations with load factor between 0.35 and 0.45. For very low load factors, 0.25 and under (as in connection with large manufacturing plants, tall office building, and other plants operating on a 12-hour basis), this efficiency seldom exceeds 50 per cent. With these figures as a guide the cost of fuel per unit output may be roughly approximated.

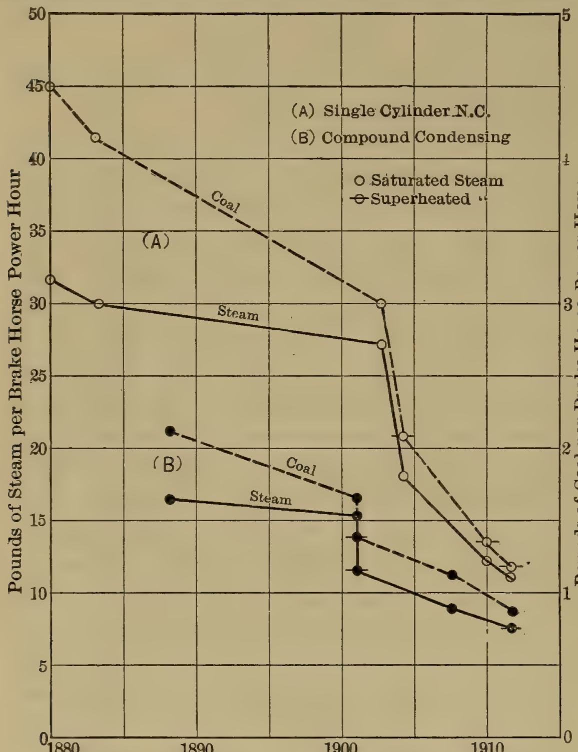


Fig. 582. Development of the Steam Power Plant. (Locomobile Type.)

In Europe the "locomobile" type of steam power plant has attained an extremely high degree of heat efficiency as will be seen from the curves in Fig. 582. The most economical result shown, namely 0.87 pound of coal per developed horsepower hour, is equalled only by our best gas-producer plants.

472. Oil, Waste, and Supplies. — These items approximate from 2 to 10 per cent of the total operating expenses. Tables 144 to 160 give some idea of current practice in different classes of power plants.

473. Repairs and Maintenance. — This item ordinarily refers to the cost of keeping the plant in running order over and above the cost of labor or attendance, and depends upon the age and condition of the plant and the efficiency of the employees. Tables 144 to 160 give the cost of repairs and maintenance for a wide range in power-plant practice.

474. Cost of Power. — The actual cost of producing power depends upon the geographical location of the plant, the size of apparatus, the

TABLE 145.

COST OF ONE HORSE POWER PER YEAR, SIMPLE ENGINES, NON-CONDENSING, 10-HOUR BASIS, 308 DAYS PER YEAR.

(Wm. O. Webber, Engineering Magazine, July, 1908, p. 563.)

Size of plant................horse power	20	40	60	80
Cost of plant per horse power...........	$200.00	$190.00	$180.00	$175.00
Fixed charges at 14 per cent	28.00	26.60	25.20	24.50
Coal per horse-power hour, in pounds....	12.00	10.00	9.00	8.00
Cost at $4.00 per ton	66.00	55.00	49.50	44.00
Attendance, 10-hour basis...............	30.00	20.00	15.00	13.00
Oil, waste, and supplies.................	6.00	4.00	3.00	2.60
With coal at $5.00 per ton...............	146.50	119.35	105.07	95.10
With coal at $4.50 per ton...............	138.25	112.47	98.80	89.60
With coal at $4.00 per ton...............	130.00	105.60	92.70	84.10
With coal at $3.50 per ton...............	121.75	98.72	86.51	78.60
With coal at $3.00 per ton...............	113.50	91.85	80.32	73.10
With coal at $2.50 per ton...............	105.25	84.97	74.13	67.60
With coal at $2.00 per ton...............	97.00	78.10	67.95	62.10

TABLE 145a.

COST OF ONE HORSE POWER PER YEAR, COMPOUND CONDENSING ENGINES, 10-HOUR BASIS, 308 DAYS PER YEAR.

(Wm. O. Webber, Engineering Magazine, July, 1908, p. 564.)

Size of plant.........horse power	100	200	300	400	500	600
Cost of plant per horse power....	$170.00	$146.00	$126.00	$110.00	$96.00	$85.00
Fixed charges at 14 per cent.....	23.80	24.40	17.65	15.40	13.45	11.90
Coal per horse-power hour, pounds	7.0	6.5	6.0	5.5	5.0	4.5
Cost of fuel at $4.00 per ton......	38.50	35.70	33.00	32.00	27.50	24.70
Attendance, 10-hour basis........	12.00	10.00	8.60	7.25	6.20	5.40
Oil, waste, supplies..............	2.40	2.00	1.72	1.45	1.24	1.08
Total	76.70	68.10	60.97	56.10	48.39	43.08
With coal at $5.00 per ton........	86.40	77.10	69.22	61.90	55.29	49.28
With coal at $4.50 per ton........	81.50	72.60	65.07	58.10	51.79	46.18
With coal at $4.00 per ton........	76.70	68.10	60.97	56.10	48.39	43.08
With coal at $3.50 per ton........	71.90	63.70	56.82	50.50	45.04	39.98
With coal at $3.00 per ton........	67.00	59.20	51.67	46.70	41.49	36.88
With coal at $2.50 per ton........	62.30	54.75	48.59	43.00	38.83	33.83
With coal at $2.00 per ton........	57.45	50.25	44.47	40.10	34.64	30.73
Size of plant.........horse power	700	800	900	1000	1500	2000
Cost of plant per horse power....	$76.00	$69.00	$64.00	$60.00	$58.00	$56.00
Fixed charges at 14 per cent.....	10.65	9.65	8.95	8.40	8.12	7.85
Coal per horse-power hour, pounds	4.0	3.5	3.0	2.5	2.0	1.5
Cost of fuel at $4.00 per ton......	22.00	19.20	16.50	13.75	11.00	8.25
Attendance, 10-hour basis........	4.70	4.15	3.75	3.50	3.25	3.00
Oil, waste, supplies..............	0.94	0.83	0.75	0.70	0.65	0.60
Total	38.29	33.83	29.95	26.35	23.02	19.70
With coal at $5.00 per ton........	43.79	39.73	34.05	29.80	25.77	21.75
With coal at $4.50 per ton........	41.04	36.28	32.00	28.05	24.39	20.72
With coal at $4.00 per ton........	38.29	33.83	29.95	26.35	23.02	19.70
With coal at $3.50 per ton........	35.54	31.48	27.87	24.60	21.64	18.67
With coal at $3.00 per ton........	32.79	29.03	25.80	22.90	20.27	17.65
With coal at $2.50 per ton........	30.04	27.18	23.75	21.20	18.89	16.60
With coal at $2.00 per ton........	27.29	24.23	21.70	19.47	17.52	15.57

TABLE 146.

SHOWING CAPITAL COSTS OF PLANTS INSTALLED AND ANNUAL COSTS OF POWER PER BRAKE HORSE POWER, AVERAGE WORKING CONDITIONS.

(H. Von Schon, *Engineering Magazine*, May, 1907.)

CLASS I. — Engines: Simple, Slide-Valve, Non-Condensing.
Boilers: Return Tubular.

Size of Plant, Horse Power.	Engines, Boilers, etc., Installed.	Capital Cost of Plant per Horse Power Installed.		Annual Cost of 10 Hours Power per B.H.P.	Annual Cost of 20 Hours Power per B.H.P.
		Building.	Total.		
10	$66.00	$40.00	$106.00	$91.16	$180.76
20	56.00	37.00	93.00	76.31	151.48
30	48.70	35.00	83.70	66.46	131.68
40	44.75	33.50	78.25	59.49	117.74
50	43.00	31.00	74.00	53.95	106.46

CLASS II. — Engines: Simple, Corliss, Non-Condensing.
Boilers: Return Tubular.

30	$70.70	$35.00	$105.70	$61.14	$117.70
40	62.85	33.50	96.35	55.50	107.10
50	59.00	31.00	90.00	50.70	97.73
60	56.00	30.00	86.70	47.42	91.34
80	50.00	27.50	77.50	43.86	85.41
100	44.60	25.00	69.60	40.55	79.19

CLASS III. — Engines: Compound, Corliss, Condensing:
Boilers: Return Tubular with Reserve Capacity.

100	$63.40	$28.00	$91.40	$33.18	$60.05
150	53.70	24.00	77.70	29.83	54.63
200	50.10	20.00	70.10	28.14	51.72
300	45.90	18.00	63.90	26.27	48.83
400	43.55	16.00	59.55	24.84	46.12
500	41.25	14.00	55.25	23.73	44.21
750	40.50	13.00	53.50	23.56	44.02
1000	39.00	12.00	51.00	23.26	43.71

design, conditions of loading, system of distribution, and the method of accounting. Tables 145 to 160 compiled from various sources give the detailed costs of a large number of central and isolated stations.

Table 147 and Figs. 586, 587, and 588 give the fundamental relations between the various items entering into the cost of power for various types of plants of over 30,000 kilowatts rated capacity. These data are taken from a paper presented by H. G. Stott at a meeting of the Toronto section of the American Institute of Electrical Engineers, Toronto, Ont., December, 1908. The figures have been brought up to date (June, 1912) by Mr. Stott and show what is actually being done to-day in large plants of the size stated above.

TABLE 147.

DISTRIBUTION OF MAINTENANCE AND OPERATION COSTS IN POWER PLANTS HAVING A MAXIMUM LOAD OVER 30,000 KILOWATTS.

(H. G. Stott.)

	Reciprocating Steam Plant.	Steam Turbine Plant.	Reciprocating Engines and Low-pressure Steam Turbines.	Gas Engine Plant.	Gas Engines and Steam Turbines.	Hydraulic.
Maintenance.						
1. Engine room, mechanical	2.59	0.51	1.55	5.18	2.84	0.51
2. Boiler or producer room	4.65	4.33	3.55	1.16	1.97	......
3. Coal- and ash-handling apparatus	0.58	0.54	0.44	0.29	0.29	......
4. Electrical apparatus	1.13	1.13	1.13	1.13	1.13	1.13
Operation.						
5. Coal	61.70	55.53	46.48	26.52	25.97	......
6. Water	7.20	0.65	0.61	3.60	2.16	......
7. Engine room, labor	6.75	1.36	4.06	6.76	4.06	1.36
8. Boiler or producer room, labor	7.20	6.74	5.50	1.81	3.05	......
9. Coal- and ash-handling, labor	2.28	2.13	1.75	1.14	1.14	......
10. Ash removal	1.07	0.95	0.81	0.54	0.54	......
11. Electrical labor	2.54	2.54	2.54	2.54	2.54	2.54
12. Engine room, lubrication	1.78	0.35	1.02	1.80	1.07	0.20
13. Engine room, waste, etc	0.30	0.30	0.30	0.30	0.30	0.20
14. Boiler room, lubrication, etc.	0.17	0.17	0.17	0.17	0.17	......
	67.91					
Relative operating cost, per cent	100.00	77.23	69.91	52.94	47.23	5.94
Relative investment, per cent	100.00	75.00	80.00	110.00	96.20	100.00
Probable average cost per kilowatt	125.00	93.75	100.00	137.50	120.00	125.00
Probable fixed charges	11%	11%	11%	12%	11.5%	11%

For steam-turbine plants larger than 60,000 kw. the cost per kilowatt may be reduced to $75.00.

TABLE 148.

COST OF ELECTRICAL POWER — OIL FUEL.

Charles C. Moore & Co. Costs Based on Installation at Pacific Coast Railroad Terminal Points. San Francisco, Cal.

Total Rated Capacity of Plant — Kw.	Type of Engine.	No. of Units.	Size of Units — Kw.	Total First Cost of Plant — Dollars. Does not Include Real Estate or Piping Outside of Building Walls.	Total Fixed Charges per Year — Dollars.	Total Cost of Labor per Year — Dollars.	Kw. Hours per Barrel of Oil 12-Hour Test.	100 per Cent Curve Load Factor.							
								Supplies per Kw. Hour — Cents.	Water per Kw. Hour — Cents.	Kw. Hours per Barrel of Oil.	Fixed Charges, Labor, Supplies, and Water per Kw. Hour — Cents.	Fuel per Kw. Hour at $1.00 per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel 50 Cents per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel 75 Cents per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel $1.00 per Barrel — Cents.
	24 Hours per Day Plants.														
50	High-speed tandem compound, non-condensing, D. C.	1	50	11,900	1,547	2,700	98	.060	.176	89	1.206	1.124	1.767	2.048	2.329
75	...do	1	75	15,200	1,976	2,700	101	.050	.176	92	.938	1.087	1.481	1.753	2.025
100	...do	1	100	18,100	2,353	2,700	110	.040	.169	100	.786	1.000	1.286	1.536	1.786
100	High-speed tandem compound, condensing, D. C.	1	100	20,300	2,639	2,700	116	.040	.016	105	.666	.952	1.142	1.380	1.618
150	...do	1	150	25,200	3,276	2,700	116	.035	.016	105	.506	.952	.982	1.220	1.458
300	Cross compound condensing, direct connected	1	300	44,200	5,746	3,840	175	.030	.011	159	.406	.629	.720	.877	1.035
500	...do	1	500	64,000	8,320	7,140	185	.020	.010	168	.383	.595	.681	.829	.978
750	...do	1	750	76,900	9,997	8,040	194	.020	.010	176	.305	.568	.589	.731	.873
1,000	...do	1	1,000	95,000	12,350	10,500	205	.017	.010	186	.288	.538	.557	.691	.826
1,500	...do	1	1,500	135,800	17,654	12,060	217	.017	.009	197	.252	.508	.506	.633	.760
2,500	...do	2	1,250	260,000	33,800	14,340	217	.017	.009	197	.246	.508	.500	.627	.753
5,000	...do	3	1,650	471,000	61,230	21,600	217	.015	.009	197	.213	.508	.467	.594	.721
10,000	...do	4	2,500	862,000	112,060	37,620	221	.013	.009	201	.193	.498	.442	.566	.690
20,000	Manhattan type, 4-cylinder, condensing, D. C.	4	5,000	1,480,000	192,400	71,280	225	.013	.009	205	.173	.488	.416	.538	.660

TABLE 148. — *Continued.*

20 Hours per Day Plants.

50	High-speed tandem compound, non-condensing, D. C.	1	50	11,900	1,547	2,700	98	.060	.176	88	1.400	1.136	1.968	2.252	2.536
75	...do	1	75	15,200	1,976	2,700	101	.050	.176	91	1.080	1.099	1.630	1.904	2.179
100	...do	1	100	18,100	2,353	2,700	110	.040	.169	99	.901	1.010	1.406	1.659	1.911
100	High-speed tandem compound, condensing, D. C.	1	100	20,200	2,639	2,700	116	.040	.016	105	.787	.952	1.264	1.502	1.740
150	...do	1	150	25,200	3,276	2,700	116	.035	.016	105	.597	.952	1.073	1.311	1.549
300	Cross compound, condensing, direct connected	1	300	44,200	5,746	2,760	175	.030	.011	158	.429	.633	.746	.904	1.062
500	...do	1	500	64,000	8,320	4,260	185	.020	.010	167	.375	.599	.674	.824	.973
750	...do	1	750	76,900	9,997	5,220	194	.020	.010	175	.308	.571	.594	.737	.879
1,000	...do	1	1,000	95,000	12,350	6,960	205	.017	.010	185	.292	.541	.562	.697	.832
1,500	...do	1	1,500	135,800	17,654	9,300	217	.017	.009	195	.272	.513	.529	.657	.785
2,500	...do	2	1,250	260,000	33,800	13,200	217	.017	.009	195	.284	.513	.540	.668	.796
5,000	...do	3	1,650	471,000	61,230	20,520	217	.015	.009	195	.248	.513	.504	.633	.761
10,000	...do	4	2,500	862,000	112,060	37,560	221	.013	.009	199	.227	.503	.478	.604	.730
20,000	Manhattan type, 4-cylinder, condensing, D. C.	4	5,000	1,480,000	192,400	71,280	225	.013	.009	203	.203	.493	.449	.572	.695

10 Hours per Day Plants.

50	High-speed tandem compound, non-condensing, D. C.	1	50	11,900	1,547	1,800	98	.060	.176	82	2.070	1.220	2.680	2.985	3.290
75	...do	1	75	15,200	1,976	1,800	101	.050	.176	84	1.605	1.191	2.201	2.498	2.796
100	...do	1	100	18,100	2,353	1,800	110	.040	.169	92	1.347	1.087	1.890	2.162	2.434
100	High-speed tandem compound, condensing, D. C.	1	100	20,300	2,639	1,800	116	.040	.016	97	1.272	1.031	1.788	2.045	2.303
150	...do	1	150	25,200	3,276	1,800	116	.035	.016	97	.978	1.031	1.494	1.751	2.009
300	Cross compound, condensing, direct connected	1	300	44,200	5,746	4,740	175	.030	.011	146	.999	.685	1.341	1.512	1.684
500	...do	1	500	64,000	8,320	4,740	185	.020	.010	154	.746	.649	1.070	1.233	1.395
750	...do	1	750	76,900	9,997	5,580	194	.020	.010	162	.599	.617	.908	1.062	1.216
1,000	...do	1	1,000	95,000	12,350	5,580	205	.017	.010	171	.518	.585	.811	.957	1.103
1,500	...do	1	1,500	135,800	17,654	7,860	217	.017	.009	181	.492	.553	.768	.906	1.045
2,500	...do	2	1,250	260,000	33,800	7,860	217	.017	.009	181	.483	.553	.759	.897	1.035

1. The above figures are approximate only and are subject to considerable variation, depending on operating conditions, etc.; first cost depending on location, freight rates, etc.
2. All plants are without economizers or superheaters.
3. Table is based on dry saturated steam at 150 pounds gauge pressure at boiler.
4. All auxiliaries are assumed steam driven. In all cases the circulating pumps are assumed to do 1,000 foot-pounds of work per pound of steam on main engine, which is the equivalent of pumping 50 pounds of water against a 20-foot head. All plants located at tide water. Cooling towers increase first cost and decrease economy.

TABLE 148. — (*Continued*).

Total Rated Capacity of Plant — Kw.	Type of Engine.	50 per Cent Curve Load Factor.						33⅓ per Cent Curve Load Factor.					
		Kw. Hours per Barrel of Oil.	Fixed Charges, Labor, Supplies, and Water per Kw. Hour — Cents.	Fuel per Kw. Hour at $1.00 per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel 50 Cents per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel 75 Cents per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel $1.00 per Barrel — Cents.	Kw. Hours per Barrel of Oil.	Fixed Charges, Labor, Supplies, and Water per Kw. Hour — Cents.	Fuel per Kw. Hour at $1.00 per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel 50 Cents per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel 75 Cents per Barrel — Cents.	Total Expenses per Kw. Hour with Fuel $1.00 per Barrel — Cents.
	24 Hours per Day Plants.												
50	High-speed tandem compound, non-condensing, D. C.	70	2.270	1.429	2.984	3.341	3.698	56	3.320	1.786	4.212	4.659	5.105
75	. . . do.	72	1.740	1.389	2.434	2.782	3.129	58	2.528	1.724	3.391	3.822	4.253
100	. . . do.	79	1.446	1.266	2.079	2.396	2.712	63	2.094	1.587	2.888	3.285	3.682
100	High-speed tandem compound, condensing, D. C.	83	1.297	1.205	1.900	2.201	2.502	67	1.926	1.493	2.672	3.045	3.418
150	. . . do.	83	.981	1.205	1.583	1.188	2.186	67	1.453	1.493	2.199	2.572	2.946
300	Cross compound, condensing, direct connected	125	.787	.800	1.187	1.387	1.587	101	1.166	.990	1.661	1.908	2.156
500	. . . do.	132	.748	.758	1.127	1.316	1.506	106	1.111	.943	1.583	1.819	2.055
750	. . . do.	139	.591	.719	.951	1.131	1.310	111	.876	.901	1.326	1.552	1.777
1,000	. . . do.	146	.560	.685	.902	1.073	1.244	118	.830	.848	1.253	1.465	1.677
1,500	. . . do.	155	.489	.645	.811	.973	1.134	125	.724	.800	1.124	1.324	1.524
2,500	. . . do.	164	.474	.610	.779	.931	1.084	154	.696	.649	1.021	1.183	1.346
5,000	. . . do.	172	.408	.581	.699	.844	.990	166	.599	.602	.900	1.051	1.201
10,000	. . . do.	178	.369	.562	.650	.790	.931	169	.541	.592	.837	.985	1.133
20,000	Manhattan type, 4-cylinder, condensing, D. C	181	.328	.553	.605	.743	.881	173	.480	.578	.769	.914	1.058

TABLE 148. — *Continued.*

20 Hours per Day Plants.

50	High-speed tandem compound, non-condensing, D. C.	69	2.662	1.449	3.387	3.749	4.112	55	3.908	1.818	4.818	5.272	5.727
75	...do	71	2.029	1.409	2.733	3.085	3.438	57	2.962	1.754	3.839	4.278	4.717
100	...do	77	1.681	1.299	2.331	2.655	2.980	62	2.447	1.613	3.253	3.656	4.059
100	High-speed tandem compound, condensing, D. C.	82	1.542	1.220	2.152	2.457	2.762	66	2.293	1.515	3.051	3.430	3.808
150	...do	82	1.164	1.220	1.774	2.079	2.384	66	1.728	1.515	2.485	2.864	3.243
300	Cross compound, condensing, direct connected	123	.835	.813	1.242	1.445	1.648	99	1.238	1.010	1.743	1.995	2.248
500	...do	130	.732	.769	1.117	1.309	1.501	105	1.087	.952	1.563	1.801	2.039
750	...do	137	.599	.730	.964	1.145	1.328	110	.887	.909	1.341	1.569	1.796
1,000	...do	144	.567	.694	.915	1.088	1.262	116	.841	.862	1.272	1.488	1.703
1,500	...do	153	.529	.654	.856	1.019	1.183	123	.785	.813	1.191	1.394	1.598
2,500	...do	162	.550	.617	.859	1.013	1.167	151	.810	.662	1.141	1.307	1.472
5,000	...do	170	.479	.588	.773	.920	1.067	162	.704	.617	1.013	1.167	1.322
10,000	...do	175	.438	.571	.723	.866	1.009	165	.644	.606	.947	1.099	1.250
20,000	Manhattan type, 4-cylinder, condensing, D. C.	179	.389	.559	.668	.808	.948	168	.571	.595	.869	1.018	1.166

10 Hours per Day Plants.

50	High-speed tandem compound, non-condensing, D. C.	63	4.036	1.587	4.830	5.227	5.623	50	5.969	2.000	6.969	7.469	7.969
75	...do	65	3.111	1.539	3.881	4.265	4.650	51	4.586	1.961	5.566	6.056	6.546
100	...do	71	2.602	1.109	3.306	3.658	4.010	56	3.827	1.786	4.720	5.167	5.613
100	High-speed tandem compound, condensing, D. C.	74	2.520	1.351	3.195	3.533	3.871	59	3.759	1.695	4.607	5.031	5.454
150	...do	74	1.934	1.351	2.610	2.947	3.285	59	2.882	1.695	3.730	4.154	4.577
300	Cross compound, condensing, direct connected	112	1.979	.893	2.426	2.649	2.872	88	2.954	1.136	3.522	3.806	4.090
500	...do	119	1.478	.840	1.898	2.108	2.318	93	2.206	1.075	2.744	3.013	3.282
750	...do	124	1.185	.807	1.588	1.790	1.991	98	1.767	1.020	2.277	2.532	2.787
1,000	...do	131	1.025	.763	1.406	1.597	1.788	104	1.527	.962	2.008	2.248	2.489
1,500	...do	139	.973	.719	1.332	1.512	1.692	110	1.450	.909	1.904	2.131	2.359
2,500	...do	147	.952	.680	1.292	1.462	1.632	132	1.413	.758	1.791	1.981	2.170

5. Water is assumed to cost 25 cents per 100 cubic feet Make-up water is assumed to be 10 per cent of total in all condensing plants. Cost of water and supplies per kilowatt hour vary with the load factor, being the least at full load.

6. Grade of oil is assumed 18,850 B.T.U. per pound, weighing 336 pounds per barrel.

7. Curve load factor is defined as ratio of average load to maximum load as shown by the average daily load curve, and must not be confounded with yearly load factor. Curve load factor is the same as yearly load factor for 24 hours per day plants only.

8. In all cases the peak load of curve is assumed corresponding to rated capacity of plant without overload.

9. The use of superheated steam would show a marked improvement in fuel consumption in any of the above plants, particularly with the lower load factors.

TABLE 149.

COST OF POWER.

Examples of Isolated Station Practice.

	Large Office Building.	Small Office Building.	Manufacturing Plant, Electro-plating.
Rated capacity ... kilowatts	500	50	500
Yearly capacity ... kilowatt hours	4,380,000	438,000	4,380,000
Actual load ... kilowatt hours	670,000	40,820	3,500,000
Yearly load factor ... per cent	17	9.1	80
Curve load factor ... per cent	17	24.7	80

Operating Charges, per Year.

Labor	$6,050.00	$1,400.00	$12,300.00
Coal and ashes	6,342.00	960.00	9,100.00
Water	642.00	75.00	....
Oil and waste	168.00	90.00	210.00
Lamps	395.00	41.00	50.00
Repairs and renewals	69.00	182.00	1,008.00
Total	$13,666.00	$2,748.00	$22,668.00

Fixed Charges, per Year.

Interest (5 per cent)	$3,500.00	$325.00	$4,500.00
Depreciation (6 per cent)	4,200.00	628.00	5,400.00
Insurance (½ per cent)	350.00	30.00	450.00
Taxes (1½ per cent)	1,050.00	90.00	1,350.00
Rental	900.00	....	....
Total	$10,000.00	$1,073.00	$11,700.00

Cost per Kilowatt Hour, Cents.

Operating charges	1.79	6.78	.65
Fixed charges	1.32	2.62	.33
Total cost	3.11	9.40	.98

TABLE 150.

COST OF POWER — ISOLATED STATIONS.

(I. D. Parsons, *Engineering Magazine*, January–February, 1902.)

Building number	1	2	3	4	5	6	7	8
Type of building	Office	Office	Office	Office	Office	Loft	Loft	Loft
Number of floors	11	Over 20	13	15	12	12	12	10
Ground plan	50×120	70×100	60×150	75×100	40×100	200×176	100×100	40×200
Number of electric elevators	4–25 H.P.	10–25 H.P.	6–27 H.P.	3–20 H.P.	8–20 H.P.	15–20 H.P.	4–20 H.P.	4–25 H.P.
Part of building lighted	All	All	All	All	6 floors	All	All	All
Type of engine	Simple	Compound	Compound	Simple	Simple	Simple	Simple	Simple
Number of generators	4	4	5	3	1	3	3	3
Total capacity of generators, kilowatts	260	575	475	300	40	300	260	90
Average load on generators	.55	.75	.75	.90	.65	.90	.75	.60
Voltage	220 & 240	120	115 & 240	116	240	240	240	240
Type of boiler	H. Tub.	W. Tub.	W. Tub.	H. Tub.	H. Tub.	H. Tub.	H. Tub.	H. Tub.
Boiler pressure	90	125	115	100	95	90	100	100
Back pressure	0	0	1 to 3	4	1 to 5	3 to 9	2 to 5	7 to 10
Kind of coal	Pea	Buck.	Pea	Buck.	Buck.	Rice	Buck.	Buck.
Capacity storage battery, ampere hours	....	3,500	....	3,000	300	....	....	....
Load factor of lighting load	.24	.43	.72	....	.87	.70	.45	....

Cost of Light, Heat, and Power.

Time covered by figures	1 year	1 year	9 months	1 year	1 year	1 year	1 year	1 year	1 year
Labor	$2,460	$7,228	$6,750	$9,000	$6,318	$1,484	$4,628	$4,776	$1,662
Coal and removal of ash	2,758	10,070	7,663	10,238	9,174	1,012	4,192	2,736	2,129
Water	254	776	756	1,004	906	97	446	204	270
Oil and waste	181	287	171	215	128	103	193	108	80
Lamps	120	600	338	450	420	44	233	33	57
Repairs and sundries	183	50	429	568	65	295	100	108	478
Interest and depreciation	1,340	10,420	3,630	4,840	5,305	1,274	2,200	2,730	908
Central station service	1,370	....	....	....	....	225	....	300	579
Total	$8,666	$29,431	$19,737	$26,315	$22,370	$4,534	$11,992	$10,995	$6,163
Output — light, kilowatt hours	91,804	409,233	395,949	550,000	....	18,123	....	....	....
Output — power, kilowatt hours	59,760	266,344	160,963	225,000	....	48,437	....	....	....
Output — total kilowatt hours	151,564	677,577	556,912	775,000	721,30	66,560	379,939	143,320	130,000
Cost per kilowatt hour	$0.057	$0.044	$0.034	$0.034	$0.031	$0.068	$0.032	$0.077	$0.047
Tons of coal consumed	919	3,007	2,250	3,012	2,019	337	1,873	863	835
Pounds coal per kilowatt hour	15.5	10.0	9.1	....	8.2	11.4	11	13.5	14.4

TABLE 151.

COST OF POWER — ISOLATED STATIONS.

(I. D. Parsons, *Engineering Magazine*, January–February, 1902.)

Building number	10	11	12	13	14	15		16	17
Type of building	Ap't H'se	Ap't H'se	Ap't H'se	Hotel	Hotel	Club		Club	Club
Number of floors	7	10	11	6	9	5		3	4
Ground plan	90×65	44×81	50×200	100×200	25×60	35×100		75×200	110×100
Number of electric elevators	1–15 H.P.	2–20 H.P.	3–25 H.P.	....	2–15 H.P.	....		1–20 H.P.	....
Floors of building lighted	All	All	All	1st & 2d	All	All		All	All
Number of generators	2	2	3	2	2	1		3	2
Total capacity of generators, kilowatts	45	69	175	66	80	20		130	50
Average load on generators	.48	.20	.60	.50	.67	.40		.40	.75
Voltage	115	240	115	118	115	116		118	118
Type of boiler	H. Tub.	Marine	W. Tube	H. Tub.	H. Tub.	H. Tub.		H. Tub.	H. Tub.
Boiler pressure	90	100	125	90	100	80		90	80
Back pressure	5	2 to 5	3 to 5	4	3 to 6	6		3 to 5	3 to 5
Kind of coal	Pea	Buck.	Buck.	Pea	Bitum.	Buck.		Buck.	Buck.
Load factor of lighting load	.43	.40	.40	.60	.67	.50		.44	.85
Cost of Light, Heat, and Power.									
Time covered by figures	1 year	6 mos.	1 year	10 mos.	1 year	5 mos. '00	5 mos. '01	1 year	1 year
Labor	$2,280	$1,320	$3,840	$480	$3,690	$325	$325	$2,700	$2,400
Coal and removal of ashes	1,995	936	5,563	620	5,530	263	389	1,355	1,551
Water	164	64	303	60	486	22	30	117	196
Oil and waste	136	72	240	92	339	6	6	56	90
Lamps	28	8	136	40	354	5	6	24	93
Repairs and sundries	45	46	200	50	500	16	16	112	90
Interest and depreciation	572	495	2,376	880	1,045	115	115	1,100	770
Central station service	....	300	....	....	....	....	....	....	584
Total	$5,220	$3,241	$12,658	$2,222	$11,944	$752	$887	$5,464	$5,774
Output — light, kilowatt hours	55,730	....	182,360	115,000	....	6,306	9,407	....	68,513
Output — power, kilowatt hours	26,400	....	50,000	....	....	....	....	....	....
Output — total kilowatt hours	82,130	26,605	232,960	232,960	115,000	6,306	9,307	93,800	68,513
Cost per kilowatt hour	$0.064	$0.120	$0.055	$0.019	$0.037	$0.12	$0.09	$0.057	$0.084
Tons of coal consumed	570	312	1,919	1,007	1,826	75	111	466	486
Pounds coal per kilowatt hour	15.5	25.1	18.4	13.5	12.4	26.7	26.4	11.1	16.6

TABLE 152.

YEARLY OPERATING COSTS IN FOUR TYPICAL CENTRAL STATIONS, STATE OF MASSACHUSETTS

Year ending June, 1909.

	Salem Electric Light Co.	Fitchburg Gas & Electric Co.	Haverhill Electric Co.	Malden Electric Co.
Type of Prime Mover	6 Engines	3 Engines	2 Turbines, 1 Engine	1 Turbine, 3 Engines
Rated station capacity, kw..	2500	2000	2300	....
Output, millions of kw. hrs...	3.106	4.006	3.721	4.715
Yearly load factor, per cent..	14.2	22.9	18.5	....
Total station operating force	14	12	13	14
Cost of fuel, dollars per ton..	4.51	4.52	3.97	3.78
Coal per kw. hr.............	3.3	3.28	3.27	3.02

Operating Costs, Cents per Kilowatt Hour.

Coal........................	0.740	0.740	0.650	0.565
Oil and waste..............	0.025	0.015	0.019	0.020
Water......................	0.027	0.025	0.003	0.045
Wages......................	0.410	0.308	0.285	0.320
Station building repairs.....	0.034	0.017	0.063	0.023
Steam equipment repairs....	0.158	0.041	0.073	0.072
Electrical equipment repairs.	0.011	0.072	0.019	0.014
Miscellaneous..............	....	0.024	0.040	0.021
Total.....................	1.412	1.242	1.152	1.08

TABLE 153.

COST OF POWER, CENTS PER KW. HOUR. STEAM-ELECTRIC CENTRAL STATIONS.

Year ending June 30, 1908.

	Boston.	Worcester.	Lowell.	Fall River.	Malden.	Cambridge.	Lynn.
Fuel.....................	.462	.703	.710	.880	.635	.690	.618
Oil and waste.............	.008	.027	.009	.032	.017	.019	.012
Water....................	.024	.034	.008	.012	.032	.055	.040
Wages....................	.192	.360	.262	.538	.342	.347	.296
Station repairs...........	.015	.012	.020	.012	.035	.021	.052
Steam repairs.............	.042	.055	.020	.037	.072	.059	.147
Electrical repairs.........	.056	.055	.009	.029	.014	.046	.045
Miscellaneous.............	.023	.000	.022	.080	.033	.000	.000
Total..................	.822	1.246	1.060	1.620	1.180	1.237	1.210
Cost of fuel per ton.......	3.99	4.79	4.75	4.68	4.49	4.40	3.60
Output, millions kilowatt hours per year..........	88.5	5.4	9.4	4.0	4.6	6.0	8.7
Capacity of station, thousands of H.P............	73.5	5.90	7.39	4.43	4.87	6.75	8.2

TABLE 154.

COST OF ONE HORSE POWER PER YEAR IN STREET-RAILROAD SERVICE FOR DIFFERENT CLASSES OF ENGINES.

1000-Horse-Power Plant. (R. C. Carpenter.)

(Sibley, *Journal of Engineering*, November, 1904, p. 92.)

	Coal per Horse Power, Pounds per Hour.	Tons per Horse Power per Year. Day—12 Hours.	Day—18 Hours.	Day—24 Hours.	Fuel Cost per Year, 18 Hours per Day, $2.00 per Ton.	Oil, Waste, and Labor.	Interest, 5 per Cent; Depreciation, 10 per Cent.	Total Yearly Cost per Horse Power.
Non-Condensing Engines.								
Simple slide-valve, average	4.63	10.14	15.21	20.28	30.42	7.30	4.44	42.16
Simple slide-valve, best	4.60	10.07	15.10	20.14	30.20	7.30	4.44	41.94
Simple Corliss, average	3.45	7.55	11.33	15.10	22.66	7.30	4.75	34.71
Simple Corliss, best	3.01	6.59	9.89	13.18	19.78	7.30	4.75	31.83
*Compound slide-valve	4.17	9.05	13.57	18.10	27.14	6.90	4.80	38.84
Condensing Engines.								
Compound slide-valve, average	3.25	7.12	10.68	14.24	21.36	6.70	4.72	32.78
Compound slide-valve, best	2.40	5.25	7.88	10.51	15.76	6.50	4.72	26.98
Compound Corliss, average	2.36	5.17	7.74	10.33	15.48	6.50	5.28	27.26
Compound Corliss, best	1.80	3.94	5.91	7.88	11.82	6.10	5.28	23.20

*The compound slide-valve engine, running non-condensing, made, in this series of tests, a poorer record than the single Corliss. This may have been due to the extremely bad conditions of loading. It is, I think, a fact that this class of engine has not been a marked success for street railway work.

TABLE 155.

OPERATING EXPENSES OF BOSTON ELEVATED RAILWAY COMPANY.

Average of All Stations at Switchboard, Oct. 25, 1912.

	Year. 1905	1906	1907	1908	1909	1910	1911	1912
Rated capacity, kilowatts	35,544	38,469	39,969	50,425	50,063	51,163	51,463	61,350
Yearly load factor	43.7	43.0	45.0	37.0	37.0	41.5	41.9	36.4
* Coal, cents per kilowatt-hour	0.45	0.47	0.55	0.56	0.45	0.48	0.44	0.41
Labor, cents per kilowatt-hour	0.17	0.17	0.18	0.21	0.20	0.17	0.17	0.17
† Repairs, cents per kilowatt-hour	....	....	....	....	....	....	....	....
Supplies, cents per kilowatt-hour	0.57	0.60	0.76	0.86	0.61	0.58	0.55	0.52
Total per kilowatt-hour	0.72	0.77	0.94	1.07	0.81	0.75	0.72	0.69
Ratio operating expense to gross earnings	0.68	0.687	0.69	0.67	0.654	0.642	0.646	....
Price of coal per ton	$3.1354	$3.1859	$3.572	$3.568	$3.209	$3.283	$3.346	$3.202

* Coal included in supplies. † Repairs included in supplies and labor.

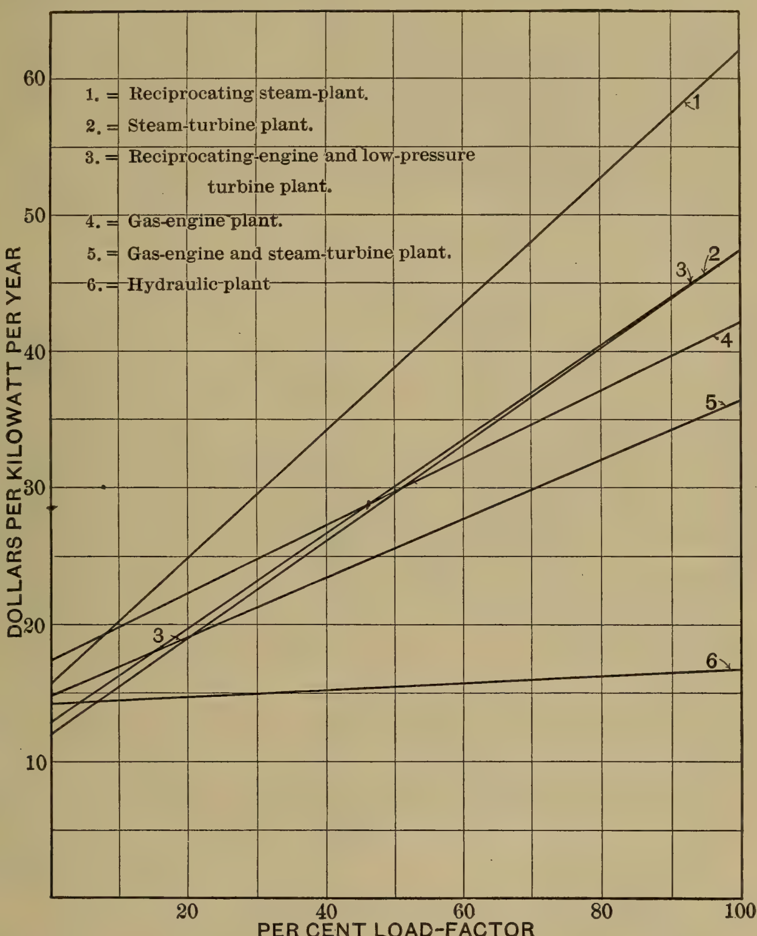


FIG. 583. Cost of Power in Large Power Plants with Maximum Load over 30,000 Kilowatts. Coal at $3.00 per Ton. 14,500 B.T.U. per Pound.

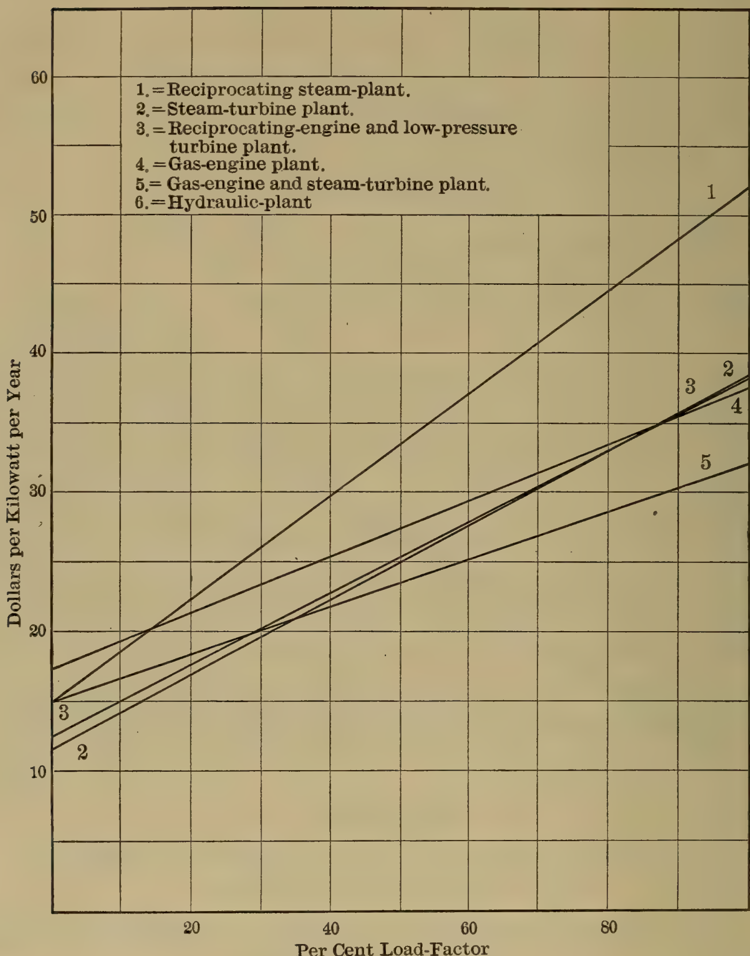


FIG. 584. Cost of Power in Large Power Plants with Maximum Load over 30,000 kilowatts. Coal at $1.50 per Ton. 11,000 B.T.U. per Pound.

TABLE 156.

COST OF POWER.

Pacific Gas and Electric Company.

Kilowatt-hours generated by steam		85,707,854
Kilowatt-hours generated by transmission		7,787,959
		93,495,813
Kilowatt-hours sold		68,797,090
Kilowatt-hours lost in distribution		24,698,723
Per cent loss, 26.5.		
TOTAL COSTS.		
Revenue from sales		$2,730,248.00
Cost of generation	$729,315.00	
Cost of distribution	347,182.00	
Cost of administration	943,363.00	2,019,860.00
Net earnings		$710,388.00

UNIT COSTS, CENTS PER KILOWATT-HOUR.

Generation:	
Labor	0.225
Materials	0.731
Repairs	0.104
	1.060
Administration:	
Labor	0.271
Materials	0.082
Legal Expenses	0.021
Fire Insurance	0.005
Bad Debts	0.026
Advertising	0.008
Damages to persons	0.005
Rental	0.005
Taxes	0.153
	0.576

Distribution:	
Labor	0.216
Materials	0.098
Repairs	0.191
	0.505
Summary of Unit Costs:	
Generation	1.060
Distribution	0.505
Administration	0.576
Interest	0.006
Depreciation	0.789
	2.936

TABLE 157.

COST OF POWER.

Year Ending June 30, 1911, Central-station Production at Fall River.

Equipment: One 500-, two 1000-kilowatt turbo-alternators.
Six 350-horse-power boilers.
Total output: 5,764,466 kilowatt-hours.
Load factor: 28 per cent.
Total Operating Costs:

Coal, 7726 tons at $3.67	$28,332.00
Oil and waste	633.00
Water	747.00
Miscellaneous supplies	598.00
Wages, 16 men	13,329.00
Repairs, building	5,537.00
Repairs, steam equipment	1,555.00
Repairs, electrical	109.00
	$51,840.00

Unit Operating Costs, Cents Per Kilowatt-hour:

Fuel	0.49
Wages	0.23
Supplies and Repairs	0.18
	0.90

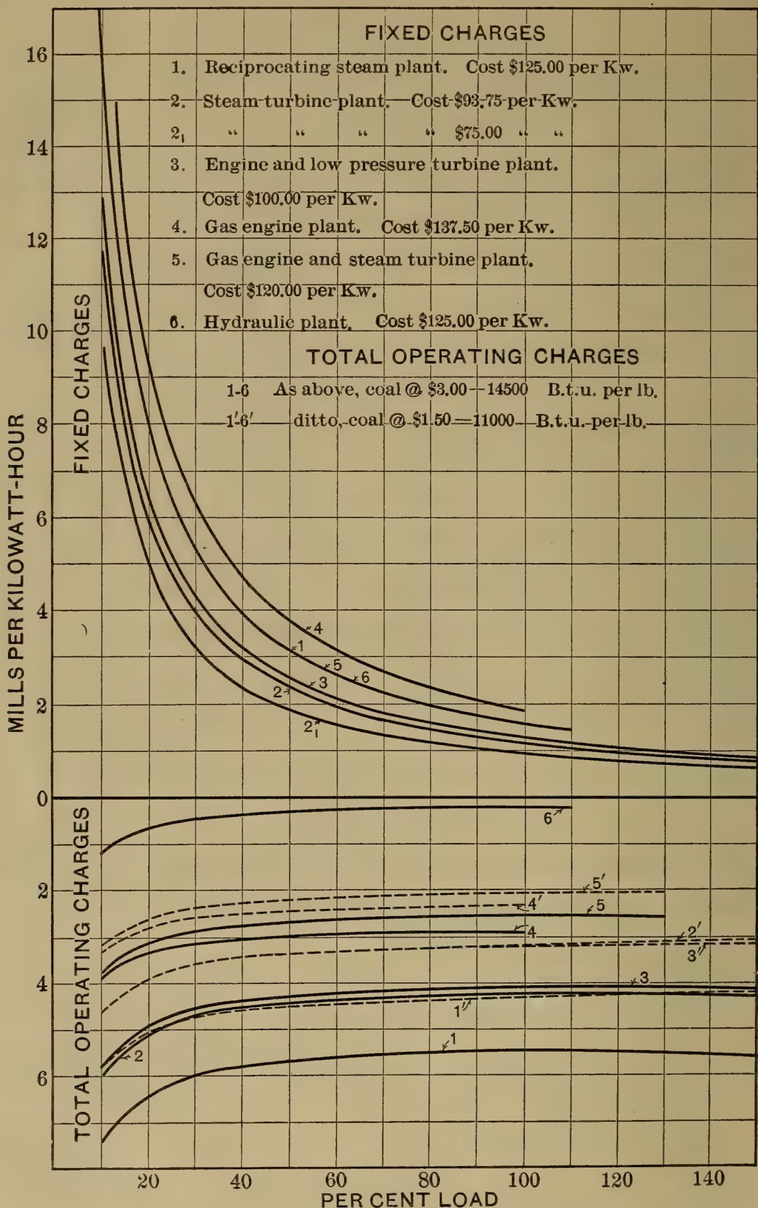


Fig. 585. Cost of Power in Large Power Plant with Maximum Load over 30,000 kilowatts.

TABLE 158.

COST OF POWER AT THE POWER PLANT OF FITCHBURG YARN COMPANY FOR THE YEAR ENDING MARCH 25, 1911.

Hewes & Phillips cross-compound Corliss engine, 26 and 56 by 60 inches:	
Boiler pressure per square inch	175 pounds.
Receiver pressure per square inch	19 pounds.
Hewes & Phillips independent-driven Venturi condenser.	
5 Dillon's Manning type boilers, 74-inch waist, 17-foot tubes:	
Grate area	192.40 square feet.
Heating surface	10,449 square feet.
Superheating surface	4662 square feet.
I. B. Davis gleaner exhaust heater:	
Water from	44 to 118 degrees
I. B. Davis gleaner heater:	
Exhaust of condensing engine and condensation from receiver, water from	118 to 152 degrees
Green fuel economizer, 4488 square feet heating surface:	
Water from	152 to 232 degrees
Gases entering economizer	430 degrees
Gases entering chimney	230 degrees
Height of chimney	165 feet
William A. Jepson coal, semi-bituminous:	
Cost of coal per ton	$4.50
Maximum indicated horse power	1291.8
Minimum indicated horse power	2107.8
Average indicated horse power	1744.28

COST OF POWER.

4229.57 pounds coal per horse power (including banking and heating)	$8.46	
Labor	2.92	
Supplies and repairs	1.11	
Total operating expenses		$12.49
Depreciation and interest	$4.01	
Taxes	0.72	
Insurance	0.04	
Total fixed charges		4.77
Total gross cost		$17.26
Deduct cost of heating		0.58
Total cost one horse power per year		$16.68

TABLE 159.

COST OF OPERATION.

POWER PLANT OF THE HYDE PARK ELECTRIC LIGHT COMPANY, BOSTON, MASS. (1910).

For the year ending June 30, 1910, total kilowatt-hours, sold	4,103,384
Of the total 88.5 per cent was for street railway service or	3,636,390
Total output at station, kilowatt-hours	4,357,648
Total generator capacity in kilowatts	1,775

Coal, bituminous at $3.94 and No. 3 Buckwheat at $3.17.

NET COST AT SWITCHBOARD.

Fuel	$34,228.04	equals $0.0078 kw.-hour
Oil and waste	746.45	
Water	707.63	
Station wages	9,609.76	equals $0.0022 kw.-hour
Repairs, station building	607.44	
Repairs, steam equipment	2,966.14	
Repairs, electric equipment	652.71	
Minor tools	776.19	
Total	$50,294.36	equals $0.0115 kw.-hour

TABLE 160.

CENTRAL-STATION STATISTICS. (STATE OF IOWA, 1909.)

CITIES 1,000 TO 2,000 POPULATION.

Station number	1	2	3	4	5	6	7	8	9	10	Av. 1–10
Population, thousands	1.1	1.2	1.4	1.5	1.5	1.5	1.8	2.0	2.0	2.0	1.6
Consumers per 100 population	12	10	9	9	7	12	10	11	11	......	10
Station Capacity:											
Kilowatts	45	50	48	100	60	60	75	120	53	60	67
Watts per capita	41	42	29	67	40	40	42	60	27	30	42
Ratio transformers to	1.2	......	......	......	1.0	......	0.1	1.7	1.1	......	1.0
Load per kw. sta. cap.:											
Lamps, kw	0.7	2.4	......	1.5	......	1.3	0.6	1.7	1.1	0.8	1.3
Motors, kw	0.0	0.1	......	0.2	......	0.2	0.0	0.0	0.0	0.0	.006
Heat and misc., kw	0.0	0.0	......	0.1	......	......	......	......	......	......	.03
Total connected, kw	0.7	2.5	......	1.8	......	1.5	0.6	1.7	1.1	0.8	1.3
Yearly load factor	32	22	46	21	......	......	50	47	......	......	36
Investment, dollars:											
Per kw. capacity	$244	$370	$250	$200	$192	$200	$300	$208	$260	$133	$236
Per capita	10	15	9	13	8	8	13	13	7	4	10
Gross annual income, dollars:											
Per kw. capacity	$81	$93	$114	$90	......	$88	$103	$70	$111	......	$94
Per consumer	26	39	43	67	......	29	42	37	27	25	37
Per capita	3.30	3.90	3.90	6.00	......	3.50	4.30	4.20	2.90	......	4.00
Ratio expense to gross income	80%	75%	65%	......	72%	76%	83%	36%	81%	64%	70%

CITIES 2,000 TO 3,000 POPULATION.

Station number	11	12	13	14	15	16	17	18	Av. 11–18
Population, thousands	2.2	2.5	2.5	2.7	2.8	3.0	3.0	3.0	2.71
Consumers per 100 population	12	6	....	11	12	9	10	10	10
Station capacity:									
Kilowatts	75	80	130	110	135	175	150	120	122
Watts per capita	34	32	52	41	48	58	50	40	44
Ratio trans. to sta. cap	0.7	....	....	1.2	0.9	1.2	D.C.	1.3	1.1
Load per kw. sta. cap.:									
Lamps, kw	2.3	1.3	2.1	1.6	2.2	1.3	1.2	1.8	1.7
Motors, kw	0.4	0.0	0.0	0.5	0.0	0.0	0.0	0.8	0.2
Heat and miscellaneous, kw	0.1	....	0.0	0.4	0.5	0.0	0.0	0.3	0.2
Total connected, kw	2.8	1.3	2.1	2.5	2.7	1.3	1.2	2.9	2.1
Yearly load factor	....	45	....	24	24	23	....	25	28
Investment, dollars:									
Per kw. capacity	$292	$187	$138	$218	$200	$200	$167	$367	$221
Per capita	10	6	7	9	9	11	8	15	9
Gross income, dollars:									
Per kw. capacity	$137	$69	....	$94	$67	$101	$67	$132	$95
Per consumer	40	40	44	36	38	65	33	50	43
Per capita	4.60	2.20	....	3.80	3.20	5.90	3.30	3.30	3.80
Ratio expense to gross income	62%	73%	72%	79%	72%	53%	....	32%	63%

CITIES 4,000 TO 53,000 POPULATION.

Station number	19	20	21	Av.	22	23	Av.	24	25	26	Av.
Population, thousands	4.0	5.0	8.0	5.67	15.0	16.0	15.5	28.0	46.0	85.0	53.0
Consumers per 100 population	11	11	9	10	7	3	5	8	3	4	5
Station capacity:											
Kilowatts	250	255	300	268	900	650	775	2100	1000	3250	2120
Watts per capita	63	51	38	51	60	41	50	75	22	41	46
Ratio trans. to sta. cap	1.2	1.8	......	1.5	0.8	0.8	0.8	0.8	1.2	0.4	0.8
Load per kw. sta. cap.:											
Lamps, kw	1.2	1.4	1.6	1.4	1.4	0.7	1.0	0.9	1.7	1.6	1.4
Motors, kw	0.9	0.3	0.3	0.5	0.5	0.4	0.5	0.6	1.0	1.1	0.9
Heat and misc., kw	0.1	0.2	......	0.15	......	0.1	0.1	0.1	0.2	0.1	0.1
Total connected, kw	2.2	1.9	1.9	2.0	1.9	1.1	1.5	1.6	2.9	2.8	2.4
Yearly load factor	23	15	21	20	16	29	22	30	17	24	24
Investment, dollars:											
Per kw. capacity	$126	$216	$266	$203	$139	......	$139	$190	$200	......	$195
Per capita	8	11	10	10	8	......	8	14	4	......	9
Gross income, dollars:											
Per kw. capacity	$106	$71	$78	$85	$75	$46	$60	$58	$109	$99	$89
Per consumer	59	33	33	42	63	57	60	56	85	97	79
Per capita	6.70	3.60	3.00	4.43	4.50	1.90	3.20	4.30	2.30	3.80	2.50
Ratio expense to gross income	43%	67%	59%	56%	53%	90%	71%	42%	57%	54%	51%

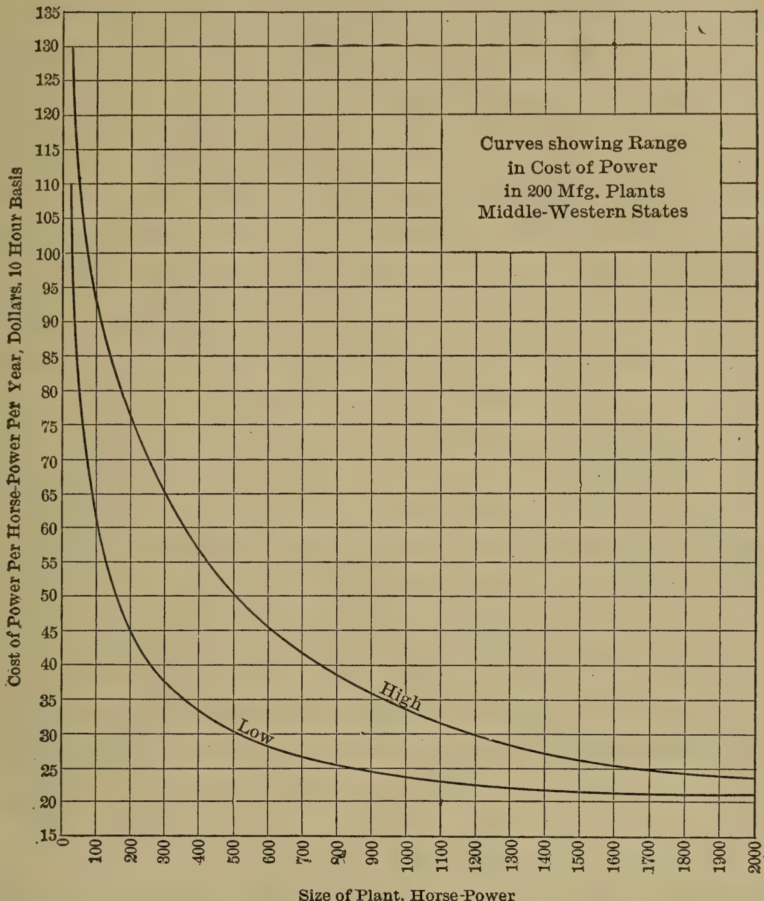


FIG. 586.

475. Elements of Power-plant Design. — The real problem which confronts the designing engineer is not so much the selection and arrangement of apparatus for a given set of conditions as it is to foresee the conditions under which the plant is likely to operate. For this reason the plans for the station should be examined and approved by an experienced designing engineer, in case expert service is not employed at the outset. It is not sufficient to have a mechanically perfect plant, though of course proper installation is of prime importance. The choice of fuel, selection of type of prime mover, size of units provision for future expansion, and similar factors bear considerable weight upon the economy of operation. Each proposed installation is

likely to be a problem in itself, and though similar plants may be used as patterns, each case should be worked out on its own merits.

The most important factor in the design of a power station is the determination of the probable load curve. This refers not only to the average yearly load but also to the maximum daily load which is likely to occur, the minimum daily load, temporary peak loads, and probable future increase. The station load factor and the yearly load factor which have such a marked bearing on the cost of operation may be closely approximated from the daily load curves. Steam requirements for heating and industrial purposes, water supply, and other forms of energy requirements should be considered simultaneously with the electrical demands since these factors largely influence the choice of

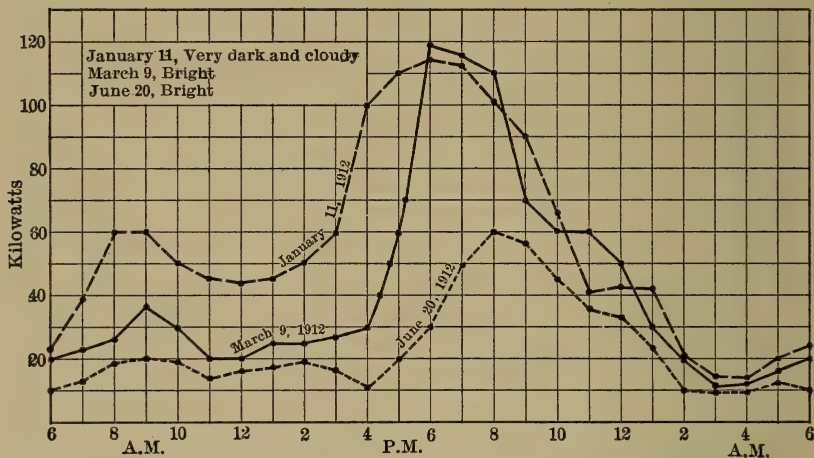


FIG. 587. Typical Daily Load Curves, Large Apartment Building.

prime mover. The curves in Figs. 587 to 589 are taken from the daily records of large power stations in Chicago and serve to illustrate the great variation in the electrical power demands for different days in the year. It is quite evident that an equipment based solely upon the average yearly requirements may not be adapted to the best economical operation.

The load curves for manufacturing plants may be predetermined with a fair degree of accuracy since the power demands for various purposes may be readily segregated and analyzed, but with public utility concerns and certain classes of isolated stations the problem is largely a matter of judgment. Thus, in the case of an industrial plant, the power requirements for lighting, manufacturing purposes, heating, ventilation, and sanitation may be closely approximated since the size of building, exposure, number of floors, and the number of

elevators afford a definite basis for analysis; but with public utility concerns the probable load depends largely upon the business acumen of the management in securing customers, the location of the plant and future demands. In the latter case the load curve is based chiefly upon the experience of similar plants under comparable conditions of operation.

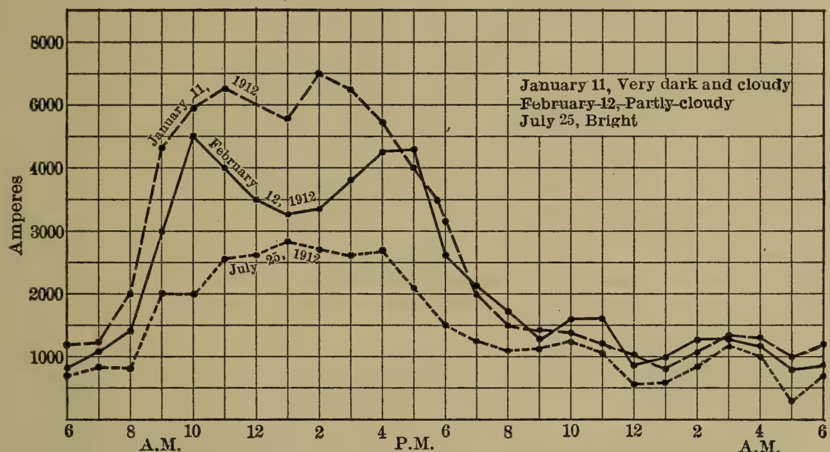


Fig. 588. Typical Daily Load Curves, Tall Office Building, Chicago.

In any case the greatest care should be exercised in estimating the maximum peak load which is likely to occur. High peak loads with low daily average necessitate the installation of large machines which are idle or operate uneconomically the greater part of the time and result in heavy fixed charges. The financial failure of many electric light and power plants is directly traceable to the failure to consider the influence of maximum peak loads on the ultimate cost of operation. In connection with central-station service every customer represents a certain investment, regardless of the amount of power used. Even should he consume no power, his account would have to be carried on the books and a certain amount of equipment would have to be held in readiness to serve him. In order that every customer shall incur his share of the expense, the expense of the plant must be apportioned between the capacity and output costs. The heavier the peak loads the greater will be this charge, and, as is the case with many small lighting plants where current is used but three or four hours a day, the readiness to serve charge becomes excessive and either the station must operate at a loss or the unit cost will appear to be prohibitive.

The curves in Fig. 590 are taken from recording ammeter and recording steam meter readings of a 200-kilowatt direct-connected and

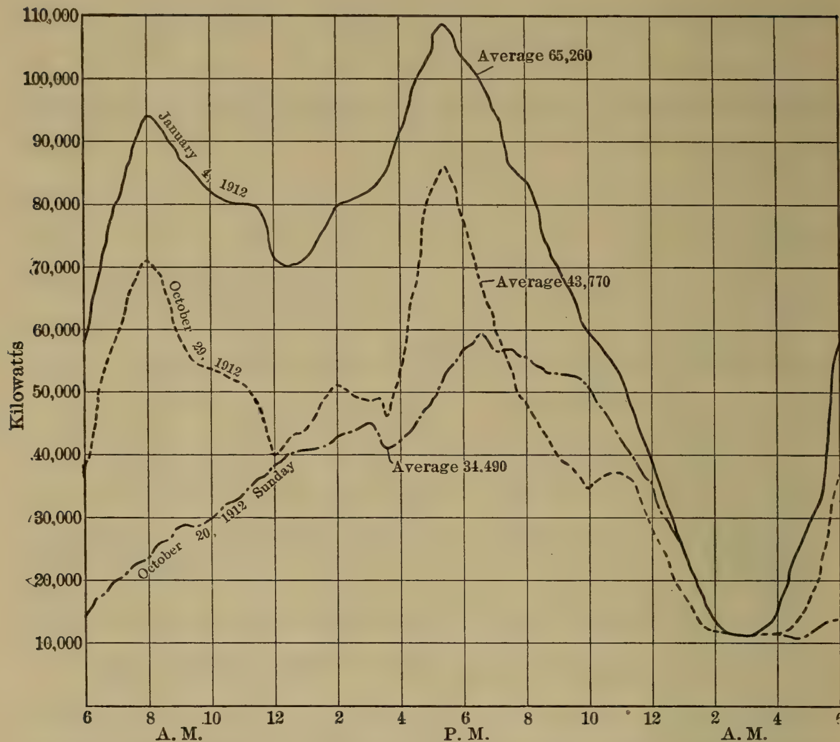


FIG. 589. Typical Daily Load Curves, Large Central Station, Chicago.

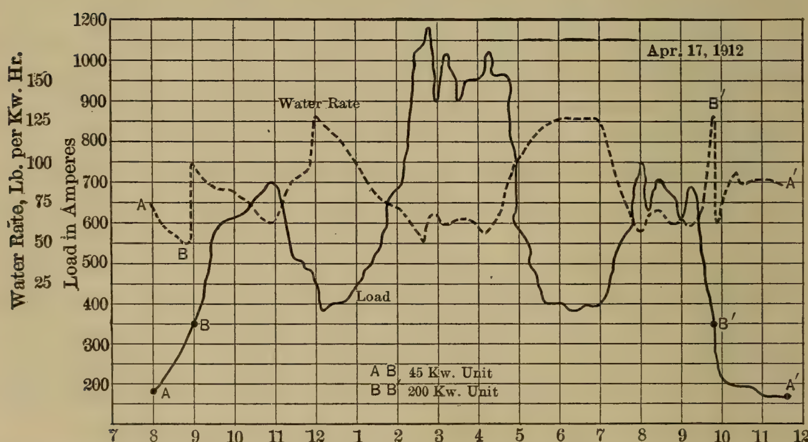


FIG. 590. Daily Load Curve showing Influence of Variable Generator Load on Steam Economy.

a 45-kilowatt belted generator set installed at the power plant of the Armour Institute of Technology and serve to illustrate the influence of load on economy for very unfavorable conditions. At 8:00 A.M. the small unit is started up with initial load of about 150 amperes. As the load increases the water rate decreases, as is shown by the curve *AB*.

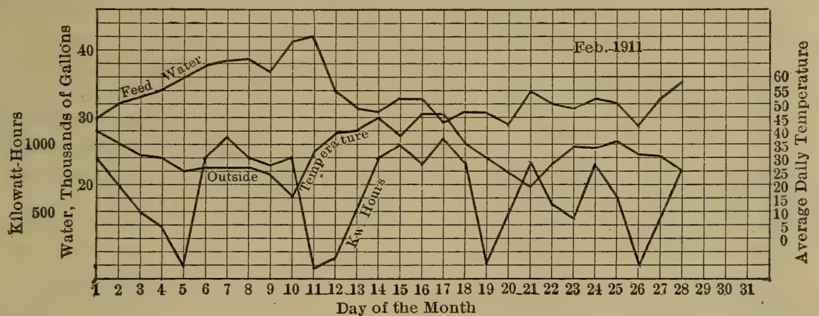


FIG. 591. Monthly Load Curves, Combined Heat and Power Plant, Armour Institute of Technology.

At 9:00 A.M. the load is beyond the capacity of the small machine and the large unit is put into service. The increased water rate of the large unit over the requirements of the smaller is apparent by the sudden rise in the water-rate curve. This is due to the fact that the large unit is operating at only 20 per cent of its rating, against full load for the

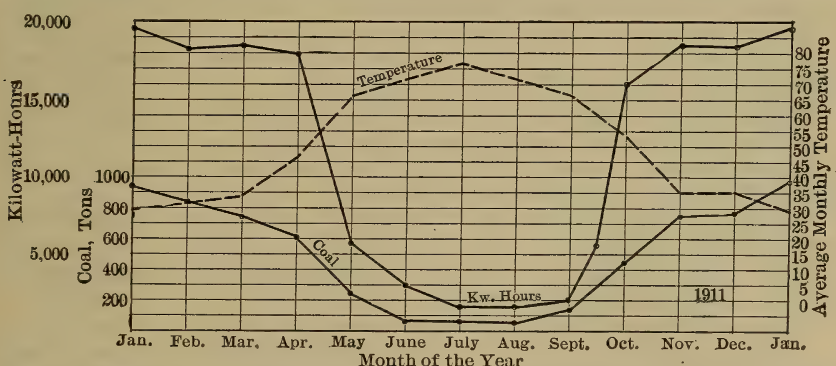


FIG. 592. Yearly Load Curve showing Influence of Temperature on Coal Consumption, Combined Heat and Power Plant, Armour Institute of Technology.

small one. The fluctuation of the water rate with the load variation is clearly shown. Evidently the two units are not of the proper size for the particular load conditions illustrated in Fig. 590. During the heating months when live steam is necessary for "make-up" purposes the unfavorable engine load has little effect on the ultimate economy,

but during the summer months the loss from this cause may be a serious one.

The curves in Figs. 590 to 592 show that during the winter months in a combined heat and power plant the fuel requirements may be practically uninfluenced by the electrical demands and increase in electrical output does not effect an appreciable increase in fuel consumption, but the influence of the outside temperature is clearly indicated.

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CHAPTER XIX.

TYPICAL SPECIFICATIONS.

476. Sample Specifications for a Cross Compound Non-Condensing Engine.—For and in consideration of the amount and terms named in the letter accompanying this specification, and of the same date, we propose to furnish f.o.b. cars at our factory, Elizabethport, N.J., for account of THE ARMOUR INSTITUTE OF TECHNOLOGY, Chicago, Ill., ONE BALL & WOOD HORIZONTAL FOUR-VALVE (CORLISS) CENTER-CRANK ENGINE, designed for direct connection to a direct-current generator, as follows:

General Dimensions.	Horse power	350 to 375
	Diameter of cylinders	H.P., 17; L.P., 27 inches
	Length of stroke	18 inches
	Revolutions per minute	175 to 200
	Governor wheel	diameter, 90; face, 21 inches
	Width of belt (if belted)	20 inches
	Diameter of steam pipe	7 inches
	Diameter of exhaust pipe	10 inches
	Crosshead pins	6 in. long, 8 in. diameter
	Crank pins	9 in. long, $9\frac{1}{2}$ in. diameter
	Main bearings	20 in. long, 9 in. diameter
	Wearing surface of crossheads	242 square inches
	Width of engine over all	14 feet 6 inches
	Length of engine over all	20 feet 3 inches
	Weight complete	60,000 pounds

Rating. The rated power of the engine specified is based on an initial pressure of 120 pounds (measured in the cylinder), cutting off at about one third stroke in both cylinders, without vacuum, when operating at 200 revolutions per minute.

Fittings. With each engine is furnished the following complete list of fittings:

- One extended shaft (omitted if belted engine),
- One sub-base with extension for dynamo (omitted if belted engine),
- One self-oiling outboard bearing (omitted if belted engine),
- One throttle valve,
- One lubricating system consisting of pipes, sight feeds. and oil reservoir,
- One cylinder lubricator, nickel plated,

Fittings — *Continued.*

One set special steel wrenches,
One socket wrench for piston,
One socket wrench for connecting rod bolts,
One steel wrench for hexagon nuts,
One connecting rod set screw wrench,
One monkey wrench,
One spanner for valve stem gland,
One eye bolt for pillow block cap,
Two push-off bolts,
One set grease cups,
Two oil cups,
One hand oil pump,
One set brass oil cups,
Three nipples for drip pipes, for frame,
Four nipples for cylinder drips, 3 inches long,
One nipple for throttle bleeder,
One globe valve for throttle bleeder,
Four globe valves for cylinder drips,
One set foundation bolts, nuts, and plates,
One template for locating bolts,
One governor wheel and keys,
One balance wheel and keys (omitted if direct connected engine),
Packing for piston and valve rods,
One one-gallon can cylinder oil,
One one-gallon can engine oil,
Two cans grease,
Two wedges for wheels,

Cylinders. Cylinders are made of hard and close-grained iron, and under the influence of oil and wear the walls will rapidly acquire a fine, smooth glaze. Radiation is prevented by a thick jacket of asbestos cement, outside of which is neatly fitted an ornamented jacket. Openings and globe valves are provided for drainage.

Connecting Rods. The connecting rods are of forged machinery steel of low carbon and fitted with heavy straps with keys and bolts for adjustment.

Crank Pin Boxes. The crank pin boxes are of cast iron lined with Babbitt metal.

Crosshead Boxes. The crosshead boxes are of the best quality of phosphor bronze.

Crossheads. The crossheads are of cast iron faced with Babbitt metal on the wearing surfaces. The crosshead pins are pressed into the crossheads.

Piston. The piston rods are of special hammered steel, threaded and screwed into the crossheads, and locked fast with special nut counterbored at the end to cover threads, finished and case-hardened. The other ends of the rods will be fitted to the pistons with thread and locked with nut. The pistons will be fitted with two rings turned eccentric and cut open at the thinnest part, the ends being halved so as to lap when in position.

Valves. Both the admission and exhaust valves are of the Corliss pattern. The former are provided with double ports, and are actuated from a wrist plate receiving its motion from the governor placed in the fly wheel of the engine. This governor controls the valves of both the high and low-pressure cylinders and possesses a range of cut-off from 0 to about $\frac{3}{4}$ stroke.

The exhaust valves are driven from a wrist plate through an adjustable eccentric by which any desired degree of compression can be obtained.

Speed. The use of Corliss valves, arranged as described in the foregoing paragraph, permits an increased speed over the common type of Corliss engine with releasing gear, and while yielding the same economy dispenses with many working parts, and, what is more important, with the large and cumbersome fly wheel which has so often proved a source of danger in slow-speed engines.

Frame. The frame is proportioned for great strength and the metal is placed where it is most needed. An oil groove is cast around the bottom of the frame to protect the foundation.

Main Bearings. Main bearings are fitted with removable Babbitt shells which can be replaced when necessary. Special care is taken to have these bearings of ample length to support the wheels and stand the strain of power transmission or the weight of armature when direct connected. In the latter case a self-oiling outboard bearing is provided to carry the outer end of shaft.

Guides. The guides are known as the locomotive pattern and are interchangeable. They are carefully scraped to surface plates and provision made for taking up wear.

Crank Shaft. The crank shaft is of the best quality of steel, being carefully counterbalanced by cast-iron disks in which the necessary weight is placed. In direct connected engines this shaft is either extended in one piece to carry the armature or made in two pieces and coupled.

Governor. The governor is of the inertia type and has a swinging eccentric, the eccentric center moving across the end of the shaft about an outside point, and giving a lead which varies with the point of

Governor — *Continued.*

cut-off from a maximum, at the latest point, to zero, when the governor weights occupy their extreme outward position. Alteration in speed is obtained by changing the amount of weight in the pockets of the lever arm.

Balance. The balance wheel (in the case of belted engine) is made with a flanged rim and with a split hub, the hub being secured to the shaft by a bolt. The other keys are square, with parallel sides, and are inserted without driving.

Oiling. The oiling system consists of a simple oil reservoir which supplies oil through a system of pipes to the points of the engine needing lubrication. After fulfilling its functions this oil is drained and can be used anew. This does away with the old cumbersome oil-cup system and has the great advantage of delivering clean oil to the engine.

Guarantees. *Material.* We guarantee that the material and workmanship are of the best and that all working parts having flat surfaces are scraped to surface plates.

Regulation. That the engine shall regulate within 2 per cent under changes of load within the range of the governor, and that no reduction of boiler pressure shall reduce the speed until the latest point of out-off is reached.

Steam Consumption. That the steam consumption, when the engine is developing its rated power at 125 pounds pressure and no vacuum, shall not exceed 22 pounds of dry steam per indicated horse power per hour; that the clearance shall not exceed 8 per cent.

Drawings. With the engine is furnished a drawing showing its details, together with foundation plans.

Preparation for Shipment. Every engine is completely erected at our works before shipment. The castings are rubbed smooth, carefully filled, and the engine given two good coats of standard shop color. All bright parts are carefully protected against corrosion. The engine is dismantled, the small parts being boxed, and in the case of export shipment the larger pieces crated.

Erection. Full drawings and directions for erecting the engine will be furnished. Template, foundation bolts, nuts, and plates to be shipped in advance if necessary, and by freight unless otherwise directed.

If requested we will furnish the services of an expert to superintend the erection of this engine at the rate of $5 per day added to his traveling expenses and board, the purchaser to furnish all laboring help.

Terms. One-half cash on presentation of bill of lading, balance on completion of erection.

The title to the apparatus herein sold shall not pass from The Ball and Wood Company until all payments hereunder (including deferred payments, if any) shall have been fully made in cash. The purchaser agrees to do all acts necessary to perfect and maintain such retention of title in the said Company. All previous communications between the parties hereto, verbal or written, are hereby abrogated and withdrawn, and this proposal, when duly signed and approved, constitutes the agreement between the parties hereto, and no modification of this accepted agreement shall be binding upon the parties hereto or either of them unless such modification shall be in writing, duly accepted by the purchaser and approved by an executive of the Company.

Limit. Prices subject to revision after thirty days. Delivery subject to strikes, accidents, or causes beyond our control.

477. Specifications for Horizontal Tubular Steam Boiler.* — The following specifications for one 54-inch horizontal return tubular steam boiler, pressure 125 pounds, were prepared by the HARTFORD STEAM BOILER INSPECTION AND INSURANCE COMPANY for the ARMOUR INSTITUTE OF TECHNOLOGY, Chicago.

The boiler to conform to the following conditions and requirements:

Type and General Dimensions. It is to be of the horizontal tubular type, set with overhanging front, and all parts and pieces are to be designed accordingly.

It is to be 17 feet 2 inches long, outside, and 54 inches in diameter, measured on the outside of the smallest ring of plates. Heads are to be 16 feet 0 inches apart, outside.

Materials: Quality, Thickness, and Tests. Shell plates are to be three-eighths of an inch thick on the edges, of open-hearth fire-box steel, having a tensile strength of not less than 55,000 pounds nor more than 62,000 pounds per square inch of section, and an elastic limit of not less than half the tensile strength, with not less than 56 per cent of ductility, as indicated by contraction of area at point of fracture under test, and by an elongation of 25 per cent in a length of 8 inches.

Heads are to be one-half of an inch thick, of best open-hearth flange steel. All plates, both of shell and heads, are to be plainly stamped with name of maker, brand, and tensile strength; brands so located that they may be seen on each plate after the boiler is finished.

Each shell plate is to bear a coupon which shall be sheared off, finished up, and tested by, or for, the maker of the boiler, at his expense. Each coupon is to fulfill the foregoing requirements as

* Drawings have been omitted.

Materials: Quality, Thickness, and Tests — *Continued.*

to strength and ductility, and stand bending down double when cold, when red hot, and after being heated and quenched in cold water, without signs of fracture. There is not to be more than 0.035 per cent of sulphur, nor more than 0.035 per cent of phosphorus in the chemical composition of the plates and heads. All plates failing to pass these tests will be rejected. All tests and inspections of material may be made at the place of manufacture prior to shipment. Certified copies of report of tests must be sent to the Hartford Steam Boiler Inspection and Insurance Company, Hartford, Conn.

Riveting. The longitudinal seams are to be of the double-riveted butt-joint type with double covering strips. They are to be arranged to come well above the fire line of the boiler, and break joints in the 3 ring courses in the usual manner. The plates are to be planed on the caulking edges before rolling.

All dimensions and proportions are to be shown on accompanying drawing No. 1502.

The girth seams are to be of the single-riveted lap-joint type; rivets to be of same size as those in longitudinal seams, and pitched $2\frac{1}{8}$ inches apart from center to center; the distance from center of rivet to the edge of the plate to be equal to $1\frac{1}{2}$ times the diameter of rivet hole.

The rivet holes are to be either drilled in place, or punched at least one-quarter of an inch less than full size; if the latter method is used, the plates, after punching, are to be rolled and bolted together, and the rivet holes drilled in place one-sixteenth of an inch larger than the diameter of the rivets. The plates are then to be disconnected. All burrs are to be removed from the edges of the holes. Should any holes be in the least out of true, they are to be brought in line with a reamer or drill; if a drift-pin is used for this purpose the boiler will be rejected.

All rivets are to be driven by hydraulic pressure, wherever possible, and allowed to cool and shrink under pressure. This pressure is to completely fill the rivet holes, producing a tight joint.

Rivet Hammer Tests. The rivets are to be of the best quality of iron or soft steel, capable of being hammered flat, when cold, to a thickness of one-half their original diameter, or when hot, to one-third their original diameter, without showing signs of fracture. In the absence of physical test, it is understood that the contractor guarantees the above quality of rivets.

Braces. There are to be 20 braces $1\frac{1}{16}$ inches in diameter in the boiler, 10 above the tubes on front head, and 10 on rear head, of the crow-foot form, arranged as shown on drawing. None of them is to be less than 3 feet 6 inches long, and each is to be fastened

Braces — *Continued.*

to shell and heads by two seven-eighths inch rivets at each end; or solid steel, diagonal braces of approved pattern, and of equal strength to the former, may be used. Care is to be exercised in setting them that they may bear uniform tension. Crow-foot braces may be flat in body, if of equal strength to those specified above.

Braces below Tubes. There are to be 4 braces below the tubes in the boiler. Two of these are to be *through* braces extending from head to head. Each brace is to be $1\frac{5}{16}$ inches in diameter, with a fork formed on rear and secured with a $1\frac{1}{8}$-inch turned bolt and nut to a crow-foot securely riveted to rear head; these are the inner or central braces. The front end of brace is to be upset to a diameter of $1\frac{5}{8}$ inches, threaded and secured to front head with a nut and washer on both the inside and outside of head.

The 2 remaining braces are each to be $1\frac{5}{16}$ inches in diameter, and secured to rear head in same manner as the through braces; the front end of the brace is to be extended forward, fitted to side of shell, and riveted there with two 1-inch rivets. All to be substantially as shown on accompanying diagram of tube head No. 2431.

Tubes: Size, Number, and Arrangement. There are to be 36 lap-welded or seamless-drawn tubes, of the best quality with regard to tensile strength and ductility. They are to be round, straight, and free from all surface defects, properly annealed on their ends, and guaranteed by the manufacturers to have been tested to at least five hundred (500) pounds per square inch internal hydrostatic pressure. Each tube is to be 4 inches in diameter, 16 feet 0 inches long, and not less than standard thickness, set in vertical rows, with a clear space between them, vertically and horizontally, of 1 inch, except the central vertical space, which is to be 2 inches, as shown on accompanying diagram of tube head No. 2431.

Holes for tubes are to be neatly chamfered off on outside. Tubes to be set with a Dudgeon expander, and beaded down at each end. Tube holes may be drilled and reamed, or may be punched one-quarter inch less than full size, then rose bitted to exact diameter.

Manholes. There are to be two manholes, one 11 x 15 inches, with pressed steel frame, double riveted to inside of shell on top, and one 10 x 15 inches, flanged in front head below tubes, with suitable plates, yokes, and bolts, the proportions of the whole such as will make them as strong as any portion of the shell of like area.

Boiler Supports. The boiler to be suspended from steel I beams, 6 inches deep, $12\frac{1}{4}$ pounds per foot, by means of eye or U bolts and plate loops. There are to be 6 loops, 2 on each side of the boiler, securely riveted to boiler shell. The I beams are to be supported on cast-

Boiler Supports — *Continued.*

iron columns of square or rectangular section 6 inches square, three-quarters inch thick. Each pair of beams is to be connected together, 3 inches apart, by tie-bolts and cast-iron separators; one separator near each end, and others at intervals of about five feet. The top and bottom flanges of columns are to be faced true.

The whole system of suspension is to be made in the best manner, properly arranged to allow free expansion of the boiler, securely held and supported in every direction, amply strong in every part, and finished complete.

Nozzles. There are to be two heavy cast nozzles, made of gun-iron or steel, one 4 inches internal diameter for steam-pipe connection, and one 6 inches internal diameter for safety-valve connection, each accurately squared on top flange, and securely riveted to boiler on top. Forged or pressed steel pipe flanges may be used in place of nozzles.

The flanges of the nozzles to correspond in diameter and thickness with standard extra heavy pipe fittings.

Smoke Opening. There is to be an opening 10 by 62 inches cut out of front connection on top for attachment of uptake or flue.

Feed Pipe. There is to be a hole tapped in front head for a brass bushing, 3 inches above the top of upper row of tubes, and 16 inches from center of boiler, on left-hand side, for $1\frac{1}{4}$-inch feed-pipe connection. The bushing is to be not less than 2 inches long, to permit both the external and internal feed pipes to be screwed into it not less than seven-eighths inch.

Also furnish and put in a $1\frac{1}{4}$-inch feed pipe extending from front head back to within two feet of rear head of boiler, thence across the boiler to near shell on right-hand side. On this end place an elbow with the outlet pointed down as shown on drawings. Feed pipe is to be properly hung from the braces.

Blow-off Pipe Connection. There is to be an extra heavy pressed steel pipe flange, riveted to bottom of shell, near rear end, and tapped to receive a 4-inch extra heavy blow-off pipe. Blow-off valve and fittings to be extra heavy.

Fusible Plug. There is to be a fusible plug in rear head, two inches above top of upper row of tubes.

Fittings. There is to be furnished one pop safety valve 3 inches in diameter, one 6-inch steam gauge, three three-quarters-inch gauge cocks, and one three-quarters-inch gauge glass 12 inches long, all to be of approved pattern, and the necessary holes to be made for their proper connection. If combination water column is used, the steam and water connections between it and the boiler must be made by pipe not less that $1\frac{1}{4}$ inches in diameter.

Castings for Setting. There is to be furnished a substantial cast-iron front, with all necessary anchor bolts, 10 feet long, closely fitting front connection doors with suitable fastening to prevent warping, closely fitting furnace doors with liner plates, rear connection door 16 x 24 inches, with liner plates, grate bars for grate, pattern to be selected by purchaser of boiler, 54 inches long by 48 inches wide, with suitable bearer bars for same, arch bars for rear connection, and all buckstaves, with the necessary bolts or tie rods, and all other castings or ironwork of any description necessary for the proper construction and setting of the boiler complete.

In General. The intent of the foregoing specification is to provide for material and workmanship of the best quality, and any details of equipment not mentioned in this specification, or not shown on the drawings, but necessary for the proper completion of the boiler ready for operation, and to be hereafter contracted for, must be of equally good quality.

The size and description of parts are to conform substantially to the details of the accompanying plan, and the boiler, complete, is to be delivered at ——— and all of the material and workmanship is to be subjected to the inspection and approval of the Hartford Steam Boiler Inspection and Insurance Company.

478. Specifications for Barometric Condenser and Auxiliaries. — The following sample specifications for a barometric condenser will give some idea of the various items called for in the purchase of a condenser and appurtenances, the italicized items being specified by the purchaser.

"We submit herewith our tender for one ——— ——— condensing plant as follows:

Rated Capacity. The condenser and auxiliary machinery will have sufficient capacity to condense *250* pounds of steam per *minute* (equivalent to the steam exhausted from engines developing *1000* horse power on a basis of *15* pounds of steam per horse power per hour) when supplied with cooling water at a temperature of *70* degrees Fahrenheit, and maintaining a vacuum at the condenser of *26* inches of mercury.

Capacity under Variable Conditions. The plant will also have to condense the quantities of steam under the varying conditions as stated below:

Steam Condensed per Minute, Pounds.	Temperature of Cooling Water, Degrees F.	Vacuum Maintained at Condenser, Inches of Mercury.
250	70	26
300	70	25
340	70	24
250	80	25
290	80	24

Quantity of Cooling Water. The volume of cooling water required when the condenser is working under the above conditions will be from *550* gallons to *650* gallons per minute.

Apparatus Furnished. The apparatus to be furnished by us will consist of:

One cast-iron condensing vessel, complete with barometric tubes and foot valves.

One automatic vacuum regulator.

A structural steel framework for supporting the condensing vessel.

One positive rotary pump for supplying the cooling water to the condenser.

One "dry" air pump.

One horizontal steam engine, arranged to drive the water pump by belt and the air pump direct, the latter placed tandem to the engine; the engine is to be fitted with a suitable governor arranged for variable speeds. Purchaser to furnish belt.

Two pulleys, one for the engine and one for the water pump.

One vacuum gauge.

Four thermometers.

We do not include any steam, air, or water pipes, valves nor foundation bolts, but will furnish plans showing suitable foundations and general arrangement of the machinery.

Price and Delivery. Our price for one barometric condensing plant, as described, including all royalties, and delivered f.o.b. cars our works, is ——————

Terms. Payments as follows: Monthly payments as the work progresses in our shops, less 10 per cent. The retained percentage to be paid when the condenser is started in service, provided this is done within a reasonable time after completion.

Superintendence. If desired, we will furnish a competent machinist to superintend the erection and starting of the plant, charging extra for his services, 50 cents per hour and his traveling and boarding expenses.

Steam Pressure. The engine driving the air and water pumps will be capable of starting and operating the plant with *100* pounds minimum steam pressure, and will be built strong enough to work under *135* pounds maximum steam pressure.

Head Pumped against. The water pump and engine driving same will be designed to raise the injection water from the cold well to the condenser, the level of the water in the cold well to be *not more than 10 feet below* the level of the water in the hot well.

Power Consumption of Auxiliaries. The engine driving the air and water pumps will require approximately 2 per cent of the main engine steam when operating under rated load and with conditions as above stated.

The rotary pump has a positive displacement (Bibus' patent)

Power Consumption of Auxiliaries — *Continued.*

and is of substantial construction. All the power for propelling the water is transmitted through the main shaft, the office of the gears being simply to keep the sealing runner in time with the propelling runner. Stuffing boxes are placed between the pump chamber and bearings to prevent any grit in the water coming into contact with the journals.

The engine and air pump is of the crank and fly wheel type, the engine being fitted with a suitable governor arranged for variable speeds, and the air pump with the ——— patented slide valve.

479. Specification for Steam, Exhaust, Water, and Condenser Piping for an Electric Power Station.* — It is intended that this specification shall cover the complete installation of steam piping, exhaust piping, injection and discharge piping, drain, drip, blow-off, and boiler-feed piping, water piping, valves, separators, anchors, fittings, etc., substantially as shown on the accompanying drawing or hereinafter described.

All the materials used throughout must be the best of their respective kinds, subject to the inspection and approval of the engineer of the purchaser.

The entire work provided for in this specification is to be constructed and finished in every part in a good, substantial, and workmanlike manner, according to the accompanying drawings and this specification to the full intent and meaning of the same, and everything necessary for the proper and complete execution of the plans and drawings, whether the same may have been herein particularly specified or not, or indicated in the plans referred to, to be done and furnished in a manner corresponding with the rest of the work as well, as truly, and as faithfully as if the same were herein particularly described and specifically provided for.

The engineer shall have full power at any time during the progress of the work to reject any materials he may deem unsuitable for the purpose for which they are intended, or which are not in strict conformity with the spirit of this specification. He shall also have the power to cause any inferior or unsafe work to be taken down and altered at the cost of the contractor.

It is to be understood that the final inspection and acceptance of the work are to take place at the building after erection, and that any inspection and acceptance of material and workmanship at the mills, shops, etc., to facilitate the progress of the work, shall not preclude rejection at the building if the same be unsuitable.

* Stevens Indicator, Vol. 18, p. 373, 1901.

Any disagreement or difference between the purchaser and the contractor, upon any matter or thing arising from this specification or the drawings which form a part thereof, shall be referred to the engineer, whose decision and interpretation of the same shall be considered final, conclusive, and binding upon both parties.

Risk and Blame. — The contractor is to assume all risks and bear any loss occasioned by neglect or accident during the progress of the work until the same shall have been completed and accepted by the engineer.

Permits. — The contractor must pay for all permits and inspectors' fees or any other charges from borough, city, county, or state officers.

Dimensions. — During the progress of the work the contractor will be required to keep at the building site a complete set of drawings and a copy of this specification. The contractor will consider dimensions shown on drawings to be approximate as to location of machinery, but sufficiently accurate for the purpose. Absolute dimensions must be gotten from the location of the boilers, engines, and other machinery after they are set. No drawing shall be scaled.

The purchaser reserves the right to put other parties at work on the premises erecting machinery and doing other work during the continuance of this contract.

The contractor must conform to such rules and regulations as are in force on the property and carry out this contract with as little interference as possible with the other work of the purchaser.

Drawings.*

The following is a list of the drawings which accompany this specification and which form a part thereof:

(Title) .... No. (Drawing Number).........
do. .... do. .... etc.

Description of Plant.

Steam Piping. — The plant consists of four (4) 250-horse-power boilers from which steam will be led to two (2) independent 16-inch steam headers. Two (2) 8-inch take-offs will be carried from each boiler, one (1) to each header. These 8-inch take-offs will consist of large radius wrought-iron bends of a quality hereinafter described. From each header a 6-inch supply will be led to the low-pressure cylinders of these engines. From the header nearest the engine room two (2) 6-inch supplies will be led through the partition wall into the

* Drawings have been omitted.

cellar for the purpose of furnishing steam to the condenser pumps. From one (1) header a 6-inch and 5-inch supply will be carried to a tandem compound engine and thence 5-inch to the two (2) exciter engines.

The condenser steam lines will be continued under the engine-room floor so as to form a reserve steam supply to the exciter engines. From the end of each 16-inch main steam header a 3-inch loop will be carried to and from the fire and boiler feed pumps. The general arrangement with sizes of pipes, separators, valves, etc., is shown on drawing No. ——.

The contractor's high-pressure steam work will begin with the steam nozzles on the boilers and end with the throttle valves of pumps and engines. The contractor will furnish all pump throttle valves but not engine throttle valves.

Condenser and Exhaust Piping. — The three (3) jet condensers and air pumps will be set under engine-room floor. Injection water will be led to the condenser through a 16-inch cast-iron pipe which begins at a point 6 feet outside the wall of the boiler house and runs from this point to the condensers. After leaving the connections for the fourth condenser, this main reduces to 14 inches in diameter; after leaving the third it reduces to 10 inches in diameter; and after leaving the second it reduces to 3 inches in diameter.

This injection water main will be connected with the city supply by a 5-inch cast-iron water pipe; and the discharge pipe will be of cast-iron and will be led from the condensers, beginning with a diameter of 5 inches, increasing to 10 inches, and again increasing to 14 inches, and then to 16 inches after the last condenser, from whence it will be carried to a point 6 feet outside of the boiler-house wall.

The low-pressure cylinder of the tandem compound engine will be connected by a 10-inch and a 12-inch pipe with No. 1 condenser. Connection will be made between both high- and low-pressure cylinders of the 750-horse-power engines with 18-inch condenser exhaust headers. Between the exhaust header and the cylinder of the engines, connections will be made with a free exhaust main leading to and from exhaust riser extending through the roof of the boiler house. The general arrangement and the sizes of the gate valves, free exhaust valves, etc., are shown on drawing No. ——.

The exhaust from the two (2) exciter engines and the three (3) condensers, the boiler feed pumps, and the fire pumps will be collected and led to a 1500-horse-power open-type feed-water heater located in the boiler room. This heater will be provided with an exhaust riser and exhaust head extending through the roof of the boiler house.

The arrangement of the auxiliary exhaust pipe is shown on drawing No.——.

Boiler Feed Water Piping. — To the feed-water heater above mentioned a 2½-inch supply of fresh water will be furnished. The boiler feed pumps will draw the feed water from the heater through a 6-inch cast-iron suction pipe. They will deliver it through 3-inch brass pipes to two (2) hot-water meters equipped with by-passes and thence through a 3-inch brass pipe line to boilers. A reserve wrought-iron 3-inch feed pipe line will be run parallel to the brass feed pipe line and so connected up to it that it may be used as a reserve in case of accident to the brass line. Connections from 3-inch lines to boilers will be 2½-inch brass pipe.

The general arrangement of the feed pipe, including gate valves, check valves, sizes, etc., is shown on drawing No. ——.

Drip Piping. — There will be two (2) systems of drip piping, including traps, trap by-passes, valves, etc.

The high-pressure system comprises all the drips from the main steam header, separators, and high-pressure steam lines. The water from these traps will be led to the feed-water heater, drawing No. ——.

The low-pressure system comprises all the drips from exhaust mains, receivers, and cylinder drain cocks.

This water will be led to a catch-basin from whence it will be pumped by a tank pump. This pump will be furnished by the purchaser, drawing No. ——.

Blow-off Pipes. — A separate and distinct wrought-iron, 2½-inch blow-off pipe will be run from the blow-off valve of each of the four (4) water-tube boilers. These pipes will be run in trenches in the boiler-room floor to catch-basins located outside of building. The purchaser will construct the trenches and catch-basins.

Water Piping. — The purchaser will make a 6-inch connection from the city main to the northwest corner of the boiler room. From this point the contractor will run 6-inch cast-iron pipe to the inlet of the fire pump. From this he will run 6-inch pipe to the several fire hydrants. He will connect this 6-inch line with the injection water main with a 5-inch cast-iron connection. Near the fire pump he will run a 2½-inch connection to the feed-water heater. The pipe around the fire pump will be so arranged that ordinarily city pressure will be maintained on the water system, but in case of need the fire pump may be used to increase this pressure.

This piping is all shown on drawing No. ——.

Wrought-Iron Pipe. — All wrought-iron pipes or steam lines to be

guaranteed full-weight lap-welded wrought-iron pipe, in accordance with the standard dimensions as given in the table below. But the 16-inch headers will be 16 inches O.D., 15.04 inches I.D., and shall weigh 87 pounds per foot.

Nominal Internal.	Actual External.	Actual Internal.	Nominal Weight per Foot.
Inches.	Inches.	Inches.	Pounds.
1	1.315	1.048	1.663
1¼	1.66	1.38	2.444
1½	1.9	1.611	2.678
2	2.375	2.067	3.609
2½	2.875	2.468	5.739
3	3.5	3.067	7.536
3½	4	3.548	9.001
4	4.5	4.026	10.665
4½	5	4.508	12.34
5	5.563	5.045	14.502
6	6.624	6.065	18.762
8	8.625	7.982	28.35
10	10.75	10.019	40.065
14	15	14.25	57.893
15	16	15.25	71.77

All pipes carrying hot water, such as feed-water, blow-offs, and high-pressure drip pipes, are to be extra heavy.

All lap-welded pipe shall be proved to 500 pounds pressure per square inch before shipment. Butt-welded pipe shall be proved to 300 pounds pressure per square inch before shipment.

Exhaust pipes are to be wrought iron or steel in every case; sizes 8 inches and above may be light O.D. tubing with flanges peened or expanded on, but must be absolutely tight at 28 inches vacuum. These pipes must be tested to 25 pounds hydrostatic pressure after erection.

All high-pressure wrought-iron pipe when in place is to be tested at 250 pounds air pressure, and must be good for a working pressure of 175 pounds. All high-pressure steam pipe to be tested after erection at 175 pounds steam pressure, to the satisfaction of the engineer in every particular.

Cast-Iron Pipe. — The cast-iron pipe for injection and discharge water and for all other purposes, as shown on the accompanying drawings, shall be made of tough gray iron not less than ⅝ inch thick at any point, free from blowholes, true to pattern, and of workman-like finish. It shall be tested at 250 pounds pressure before erection, and when in place shall be tested to 25 pounds hydrostatic pressure.

Cast-iron pipes inside of buildings are to be flanged with flanged fittings.

Outside of buildings they shall be bell and spigot ends, and must be coated with tar both inside and outside.

Valves.— All steam valves over two inches, except throttle valves, will be gate valves. All gate valves will be Chapman's best make or equivalent.

On high-pressure steam lines, valves must be fitted with bronze removable seats, outside screw and yoke, and by-passed from 5 inches up.

On exhaust lines, standard pattern gate valves will be used (Chapman or equivalent), with inside screw.

Globe valves on high-pressure steam and water lines will be Schutte's make or equivalent.

Free exhaust valves will be either Schutte's make or equivalent.

Exhaust valves between engines and condensers, injection water valves at condensers, as well as priming water valves, will have their stems extended and fitted with floor stands, so that they can be operated from floor above. Floor stands will be polished all over, with polished hand wheels and indicators.

All high-pressure valves will be tested to the satisfaction of the engineer, at a hydrostatic pressure of 350 pounds per square inch and air pressure of 100 pounds between gates.

Standard valves for exhaust and water will be tested to 150 pounds.

Flanges and Fittings. — All flanges and fittings for high-pressure steam lines 3 inches and over will be open-hearth steel castings. These castings must be perfectly solid, made from heavy patterns, and free from blowholes or other defects. They must be tested at works to 500 pounds hydrostatic pressure, and guaranteed for a working steam pressure of 175 pounds per square inch.

The diameter, thickness, and drilling of flanges must not be less than the dimensions given in the following table.

Size of Pipe.	Diameter of Flange.	Thickness.	Number of Bolts.	Size of Bolts.
Inches.	Inches.	Inches.		Inch.
16	25	$1\frac{3}{4}$	20	1
14	23	$1\frac{5}{8}$	16	1
12	20	$1\frac{1}{2}$	16	1
10	$17\frac{1}{2}$	$1\frac{3}{8}$	12	1
9	16	$1\frac{5}{16}$	12	$\frac{7}{8}$
8	15	$1\frac{1}{4}$	12	$\frac{7}{8}$
7	14	$1\frac{1}{4}$	12	$\frac{3}{4}$
6	13	$1\frac{3}{16}$	8	$\frac{7}{8}$
5	11	$1\frac{1}{8}$	8	$\frac{3}{4}$
$4\frac{1}{2}$	$10\frac{1}{2}$	$1\frac{1}{16}$	8	$\frac{3}{4}$
4	10	$1\frac{1}{16}$	8	$\frac{3}{4}$
$3\frac{1}{2}$	9	1	8	$\frac{5}{8}$
3	9	1	8	$\frac{5}{8}$

Below 3 inches, on all high-pressure steam, water, or drip lines, extra heavy screwed cast-iron fittings are to be used, with sufficient extra heavy flange unions, so that any section of pipe can be readily taken out without disturbing the balance.

The brass boiler feed piping will be made up with extra heavy brass screwed fittings made from cast-iron patterns.

Pipe will be iron pipe size.

Brass flange unions will be made from standard cast-iron patterns, and a sufficient number used to make up the sections readily.

On low-pressure piping, standard fittings and flanges are to be used. Sizes 6 inches and above are to be flanged, and below 6 inches, screwed, with sufficient number of flange unions in same.

All high-pressure flanges are to be recessed on face, pipe is to be screwed, then peened up tight, and must stand the tests given above.

Standard flanges must stand the tests as stated.

All bolt holes must be drilled in solid metal; no cored holes will be allowed.

On blow-off lines, long sweep fittings are to be used, with extra heavy flange unions.

Gaskets and Bolts. — All gaskets on high-pressure steam lines must be copper.

On exhaust and water lines metallic gaskets are to be used.

All bolts must have hexagonal heads and nuts, no matter what size, points to be finished and nuts to be cold punched.

Pipe Covering. — All high-pressure steam lines, separators, and all drip lines between high-pressure steam lines and their traps will be covered with an approved magnesia pipe covering. All fittings will have molded magnesia coverings to fit them neatly.

Exhaust Heads. — The contractor will furnish and erect two (2) exhaust heads, one on the free exhaust from the engines and the other on the free exhaust from the feed-water heater.

Pipe Supports. — There will be furnished and erected the necessary wrought-iron hangers to support the two (2) 16-inch headers in the boiler room. There will also be supplied the necessary supports for the steam mains to the pumps and exciter engines. The injection water main and the condenser exhaust header will be hung from the engine room floor beams. The free exhaust header and the discharge water main will be supported on piers and saddles from the cellar floor. The saddles to be furnished by the contractor.

Separators. — The contractor will furnish and erect, as shown on drawing No. ——, eleven (11) steam separators.

Anchors. — All live-steam mains, and especially the 16-inch steam headers in the boiler room, will be anchored so as to secure no movement (at point of anchorage) on account of the expansion and contraction or vibration. Anchors to be placed where needed after the rest of the work is completed.

Relief Valves. — The boiler-feed and fire pumps shall be fitted with automatic controlling devices so that a certain definite maximum pressure may be maintained on the feed-water and fire systems.

Overflow Pipes. — The contractor will furnish an overflow relief valve for the fire pump and will connect same with the drain outside the boiler house. He will also connect the blow-off and overflow from the feed-water heater with the said drain.

Long Radius Bends. — Where shown on the drawings, changes in direction of pipe runs will be made with long radius bends. These bends will be of wrought-iron of the quality described above. Nine inches of straight pipe will be left on the ends of the bends to cut necessary threads.

The following table shows the minimum radii for these wrought-iron bends.

Size.	Minimum Radius.	Size.	Maximum Radius.	
Inches.	Inches.	Inches.	Feet.	Inches.
2	4½	7	3	0
2½	6	8	3	4
3	8	9	4	0
3½	10	10	4	4
4	14	12	5	6
4½	16	14	7	0
5	20	16	7	6
6	24	....	....	

I-Beam Supports. — The contractor will furnish and erect on the top chord of the roof trusses in the boiler room a sufficient length of I beams or channels from which he will hang the 16-inch headers. The steam connections between 16-inch header and the engines will be supported by underneath rod trusses, and vibration will be taken up by means of lateral ties.

Flange Covering. — After all the other work is completed and the plant has been in operation not less than two (2) weeks, the contractor will cover all flanges and other joints with an approved magnesia covering.

Reducing and Relief Valves. — The contractor will furnish and erect two (2) reducing valves on the throttle valves of the low-pressure cylinders of the 750-horse-power engines. These valves will be of the Kiely type or equivalent.

There will also be furnished and erected the necessary traps from the receivers between the high and low-pressure cylinders.

Check Valves. — All drip lines will be equipped with check valves which will be set at the lowest point in the line. Two (2) three-inch brass checks will be placed on the boiler feed pump outlets, all as shown on drawing No.——.

The outlet of the boiler feed pumps will also have a relief valve which will return the water to the feed-water heater in case the pressure on the pump outlet should rise above a certain definite point.

Traps. — All traps on high-pressure drip lines will be extra heavy steam traps.

Painting. — All pipes shall be painted with two (2) coats of slate graphite and linseed oil, or other good pipe paint satisfactory to the engineer.

The exhaust and condenser piping must be thoroughly painted twice under vacuum.

Damper Regulator. — The purchaser will furnish the damper regulator, but the contractor will set it and connect it with both steam headers, water supply, and damper.

Shields. — Where steam mains pass through the partition wall of the building the contractor will neatly close the opening with a sheet-metal shield so as to prevent dust from the boiler room from entering the engine room.

Meters. — The contractor will furnish and connect as shown on drawing No. —— two (2) 3-inch hot-water meters, approved by the engineer, to have a capacity of 200 gallons per minute.

Gauges. — The purchaser will furnish the customary gauges and boards, which will be set up and connected by the contractor.

Machinery. — The purchaser will furnish all engines, pumps, condensers, feed-water heater, boilers, foundations, but the contractor shall connect up the above machinery according to the evident intent and meaning of this specification.

The purchaser will not furnish any pipe, valves, fittings, pipe covering, etc., or anything connected with the work except the machinery mentioned above.

480. Government Specification and Proposal for Supplying Coal.

U. S. Treasury Department.

United States.................................

..................................., 190..

PROPOSAL.

Sealed proposals will be received at this office until 2 o'clock p. m.,
........................, 190.., for supplying coal to the United States
....building at
....as follows:
....
....
....

The quantity of coal stated above is based upon the previous annual consumption, and proposals must be made upon the basis of a delivery of 10 per cent more or less than this amount, subject to the actual requirements of the service

Proposals must be made on this form, and include all expenses incident to the delivery and stowage of the coal, which must be delivered in such quantities, and at such times within the fiscal year ending June 30, 190 , as may be required.

Proposals must be accompanied by a deposit (certified check, when practicable, in favor of ....) amounting to 10 per cent of the aggregate amount of the bid submitted, as a guaranty that it is bona fide. Deposits will be returned to unsuccessful bidders immediately after award has been made, but the deposit of the successful bidder will be retained until after the coal shall have been delivered, and final settlement made therefor, as security for the faithful performance of the terms of the contract, with the understanding that the whole or a part thereof may be used to liquidate the value of any deficiencies in quality or delivery that may arise under the terms of the contract.

When the amount of the contract exceeds $10,000, a bond may be executed in the sum of 25 per cent of the contract amount, and in this case, the deposit or certified check submitted with the proposal will be returned after approval of the bond.

The bids will be opened in the presence of the bidders, their representatives, or such of them as may attend, at the time and place above specified.

In determining the award of the contract, consideration will be given to the quality of the coal offered by the bidder, as well as the price per ton, and should it appear to be to the best interests of the Government to award the contract for supplying coal at a price higher than that named in lower bid or bids received, the award will be so made.

The right to reject any or all bids and to waive defects is expressly reserved by the Government.

DESCRIPTION OF COAL DESIRED.*

Bids are desired on coal described as follows:

....

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....

Coals containing more than the following percentages, based upon dry coal, will not be considered:

Ash.. ......per cent.

Volatile matter.............................. ......per cent.

Sulphur...................................... ......per cent.

† Dust and fine coal as delivered at point of consumption ......per cent.

DELIVERY.

The coal shall be delivered in such quantities and at such times as the Government may direct.

In this connection, it may be stated that all the available storage capacity of the coal bunkers will be placed at the disposal of the contractor to facilitate delivery of coal under favorable conditions.

After verbal or written notice has been given to deliver coal under this contract, a further notice may be served in writing upon the contractor to make delivery of the coal so ordered within twenty-four hours after receipt of said second notice.

Should the contractor, for any reason, fail to comply with the second request the Government will be at liberty to buy coal in the open market, and to charge against the contractor any excess in price of coal so purchased over the contract price.

SAMPLING.

Samples of the coal delivered will be taken by a representative of the Government.

In all cases where it is practicable, the coal will be sampled at the time

* NOTE.—This information will be given by the Government as may be determined by boiler and furnace equipment, operating conditions, and the local market.

† NOTE.—All coal which will pass through a $\frac{1}{4}$-inch round-hole screen.

it is being delivered to the building. In case of small deliveries, it may be necessary to take these samples from the yards or bins. The sample taken will in no case be less than the total of one hundred (100) pounds, to be selected proportionally from the lumps and fine coal, in order that it will in every respect truly represent the quality of coal under consideration.

In order to minimize the loss in the original moisture content the gross sample will be pulverized as rapidly as possible until none of the fragments exceed ½ inch in diameter. The fine coal will then be mixed thoroughly and divided into four equal parts. Opposite quarters will be thrown out, and the remaining portions thoroughly mixed and again quartered, throwing out opposite quarters as before. This process will be continued as rapidly as possible until the final sample is reduced to such amount that all of the final sample thus obtained will be contained in the shipping can or jar and sealed air-tight.

The sample will then be forwarded to the Chief Clerk of the Treasury Department, care of the storekeeper.

If desired by the coal contractor, permission will be given to him, or his representative, to be present and witness the quartering and preparation of the final sample to be forwarded to the Government laboratories.

Immediately on receipt of the sample, it will be analyzed and tested by the Government, following the method adopted by the American Chemical Society, and using a bomb calorimeter. A copy of the result will be mailed to the contractor upon the completion thereof.

CAUSES FOR REJECTION.

A contract entered into under the terms of this specification shall not be binding if, as the result of a practical service test of reasonable duration, the coal fails to give satisfactory results due to excessive clinkering, or to a prohibitive amount of smoke.

It is understood that the coal delivered during the year will be of the same character as that specified by the contractor. It should, therefore, be supplied, as nearly as possible, from the same mine or group of mines.

Coal containing percentages of volatile matter, sulphur, and dust higher than the limits indicated on line 54, and coal containing a percentage of ash in excess of the maximum limits indicated in the following table will be subject to rejection.

In the case of coal which has been delivered and used for trial, or which has been consumed or remains on the premises at the time of the determination of its quality, payment will be made therefor at a reduced price computed under the terms of this specification.

Occasional deliveries containing ash up to the percentage indicated in the column of "Maximum limits for ash," on page 700, may be accepted.

Frequent or continued failure to maintain the standard established by the contractor, however, will be considered sufficient cause for cancellation of the contract.

* PRICE AND PAYMENT.

Payment will be made on the basis of the price named in the proposal for the coal specified therein, corrected for variations in heating value and ash, as shown by analysis, above and below the standard established by contractor in this proposal. For example, if the coal contains two (2) per cent, more or less, British thermal units than the established standard, the price will be increased or decreased two (2) per cent accordingly.

The price will also be further corrected for the percentages of ash. For all coal which by analysis contains less ash than that established in this proposal a premium of 1 cent per ton for each whole per cent less ash will be paid. An increase in the ash content of two (2) per cent over the standard established by contractor will be tolerated without exacting a penalty for the excess of ash. When such excess exceeds two (2) per cent above the standard established, deductions will be made from price paid per ton in accordance with following table:

Ash as established in proposal.	No deduction for limits below.	Cents per ton to be deducted.							Maximum limits for ash.
		2	4	7	12	18	25	35	
		Percentages of ash in dry coal.							
Per cent.									
5	7	7–8	8–9	9–10	10–11	11–12	12–13	13–14	12
6	8	8–9	9–10	10–11	11–12	12–13	13–14	14–15	13
7	9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	14
8	10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	14
9	11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	15
10	12	12–13	13–14	14–15	15–16	16–17	17–18	....	16
11	13	13–14	14–15	15–16	16–17	17–18	18–19	....	16
12	14	14–15	15–16	16–17	17–18	18–19	19–20	....	17
13	15	15–16	16–17	17–18	18–19	19–20	20–21	....	18
14	16	16–17	17–18	18–19	19–20	20–21	21–22	....	19
15	17	17–18	18–19	19–20	20–21	21–22	....	....	19
16	18	18–19	19–20	20–21	21–22	22–23	....	....	20
17	19	19–20	20–21	21–22	22–23	....	....	....	21
18	20	20–21	21–22	22–23	....	....	....	....	22

* NOTE.—The economic value of a fuel is affected by the actual amount of combustible matter it contains, as determined by its heating value shown in British thermal units per pound of fuel, and also by other factors, among which is its ash content. The ash content not only lowers the heating value and decreases the capacity of the furnace, but also materially increases the cost of handling the coal, the labor of firing, and the cost of the removal of ashes, etc.

Proposals to receive consideration must be submitted upon this form and contain all of the information requested.

. . . .

. . . ., 190

The undersigned hereby agree to furnish to the U. S. . . . . building at . . . ., the coal described, in tons of 2,240 pounds each and in quantity, 10 per cent more or less than that stated on page 697, as may be required during the fiscal year ending June 30, 190 , in strict accordance with this specification; the coal to be delivered in such quantities and at such times as the Government may direct.

Price per ton (2,240 pounds) . . . . $ . . . .
Commercial name of the coal . . . .
Name of the mine or mines . . . .
Location of the mine or mines . . . .
Name or other designation of the coal bed or vein . . . .
Size (indicate information which will apply) —

Unsized . . . . Lump . . . . Run of mine . . . .

Screened, through . . . . . inch and over . . . . . inch { Round } Openings. / { Square } Openings. / Bar screen.

Data to establish a basis for payment:

British thermal units in coal as delivered . . . .

Ash in dry coal (Method of American Chemical Society) . . . . per cent.

It is important that the above information does not establish a higher standard than can be actually maintained under the terms of the contact; and in this connection it should be noted that the small samples taken from the mine are invariably of higher quality than the coal actually delivered therefrom. It is evident, therefore, that it will be to the best interests of the contractor to furnish a correct description with average values of the coal offered, as a failure to maintain the standard established by contractor will result in deductions from the contract price, and may cause a cancellation of the contract, while deliveries of a coal of higher grade than quoted will be paid for at an increased price.

Signature: . . . .
Address: . . . .

Name of corporation, ...
Name of president, ...
Name of secretary, ...
Under what law (State) corporation is organized: . . . .

CHAPTER XX.

TYPICAL CENTRAL STATIONS.

Steam Turbines. — Alternating Currents.

Fisk Street Station. — The Fisk Street Station of the Commonwealth Edison Company, Chicago, is an excellent example of modern central-station practice. The present (November, 1912) rated capacity of the plant is 120,000 kilowatts, though space is available for a considerable increase. The station is located on the banks of the Chicago River near Fisk and Twenty-second streets, as indicated in Fig. 593, and is about one and one-half miles south of the center of distribution of the present load. The location of the station between the east and west slips of the river secures an unusual advantage in the location of the intake and discharge tunnels, and the extent of property affords ample storage capacity for coal. The Chicago, Burlington & Quincy railway extends into the property, giving excellent facilities for the transportation of coal, ashes, construction materials, and machinery. The plant is constructed on the unit basis, each turbine and generator having its own boilers, auxiliaries, and piping system, thus permitting any unit to be shut down without interfering with the operation of the rest of the system.

Building. — The main building rests on piles, driven to hard pan, capped with a grillage of I-beams and concrete. The walls are of red pressed brick trimmed with white Bedford stone. The windows are 25 feet wide and 32 feet high, the sections of which are operated by compressed air. Fig. 593a gives a view of the north elevation. Large skylights afford ample light and ventilation. The entire interior wall surface of the turbine room is finished with white enameled brick trimmed with terra cotta. The boiler-room walls have an eight-foot wainscoting of enameled brick, the remainder being red pressed brick. The floors are of concrete, that in the turbine room being covered with two-inch hexagonal terra-cotta tile. The roofs are of Roebling concrete. The total width of the building is 243 feet, the turbine room taking up 61 feet, the boiler room 142 feet, and a car track the remainder. Two motor-driven cranes span the turbine room and run the full length of the building. A five-ton auxiliary hoist is also provided on the main

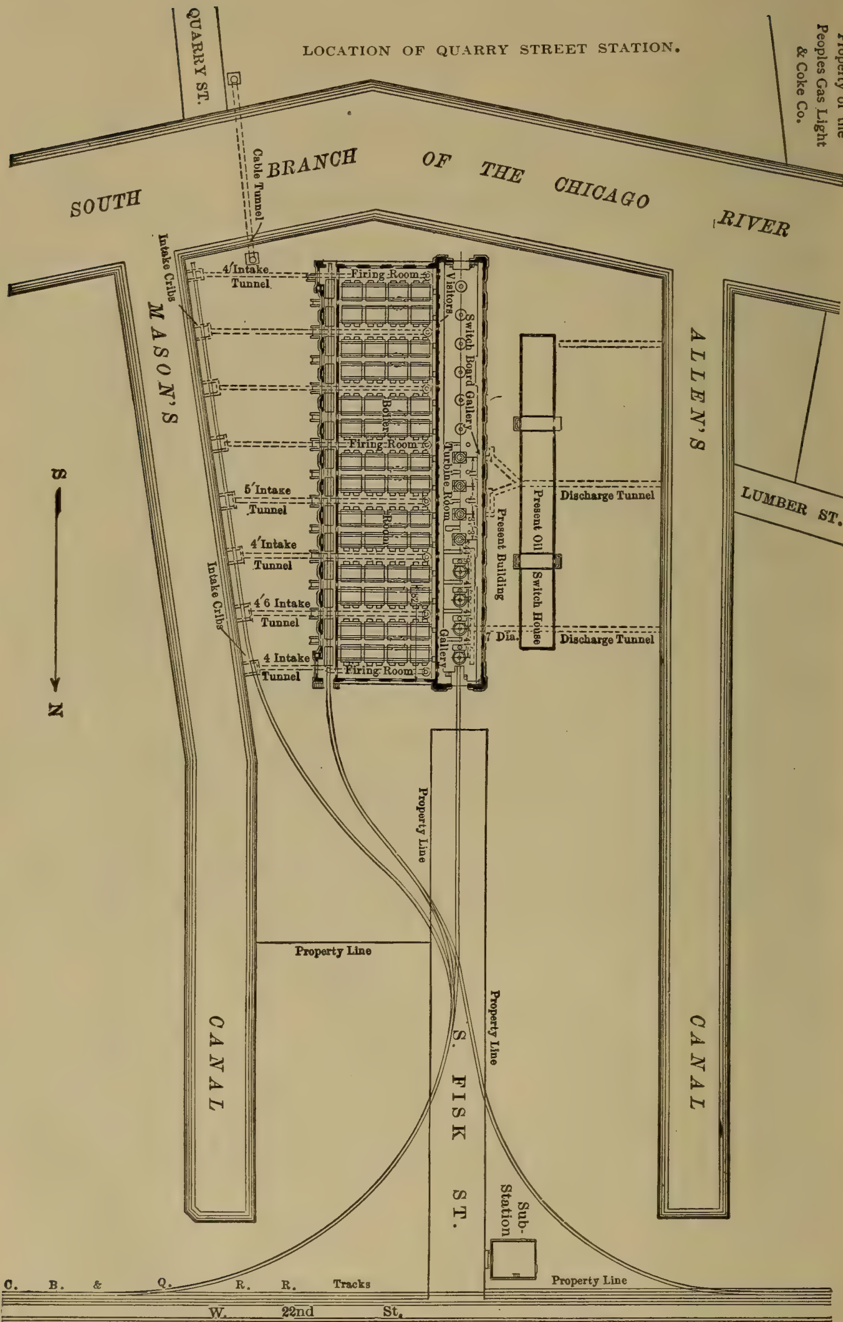


FIG. 593. Fisk Street Station of the Commonwealth-Edison Company, Chicago, Ill.

cranes. In the boiler room a small hand-power crane serves each two batteries of boilers.

Coal and Ash Handling. — An interior shed extends the entire length of the east end of the building, as indicated in Figs. 594 and 595. Coal is brought in on cars and dumped or shoveled into a track hopper, from which it is delivered to the overhead bunkers by the conveying system. A crusher is placed between the track hopper and main conveyor to be used in case lump coal is furnished. These bunkers have a capacity of 1200 tons each, sufficient for several days run. The conveyors are driven by a 15-horse-power motor and are of the McCaslin pattern, endless chain, with overlapping buckets, each bucket having a capacity of 100 pounds. The conveyors move at a variable speed giving a service capacity up to 75 tons per hour for each unit. A separate conveyor and bunker is installed in each section of 8 boilers. The coal bunkers feed through flexible down spouts to the stoker magazines. Underneath the front end of the stoker is a fine-coal hopper which collects the fine coal falling through the grate and discharges it into the conveyor system, as in Fig. 97. The ashes collect in the ash pit, from which they are dumped into the conveyor and carried to an ash bin directly over the coal track. Illinois screenings furnish the greater part of the fuel. Provision is also made for outside storage.

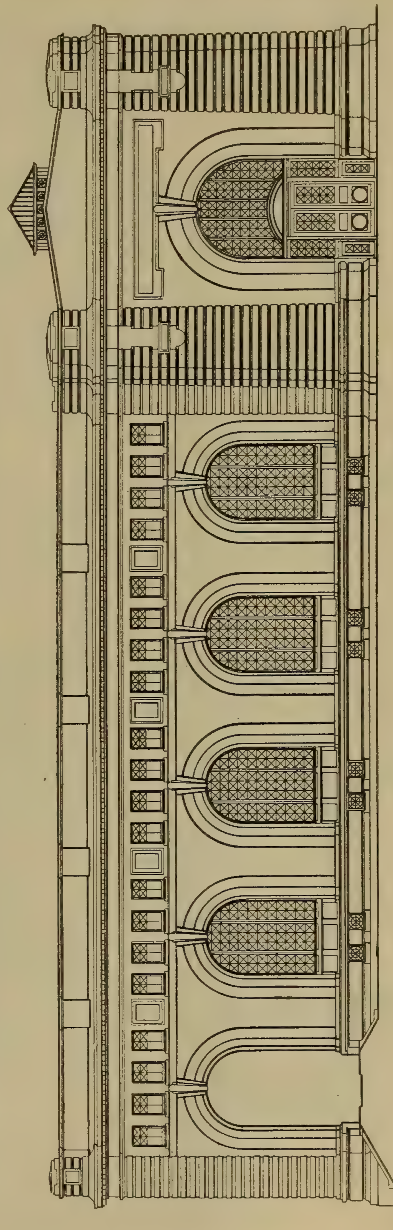


FIG. 593a. North Elevation of Building.

Boilers. — The boiler plant is divided into five sections, each section

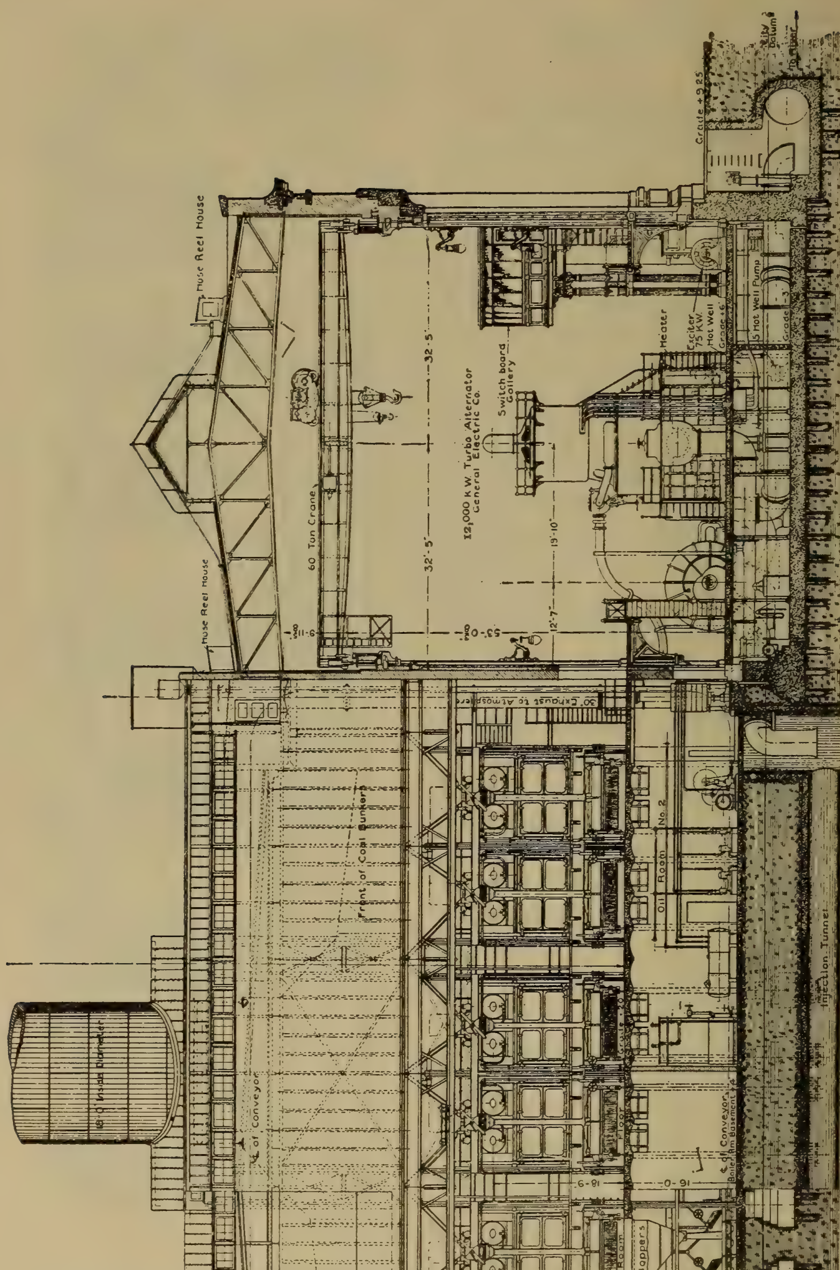


FIG. 594. Side Elevation Unit No. 10, Fisk Street Station.

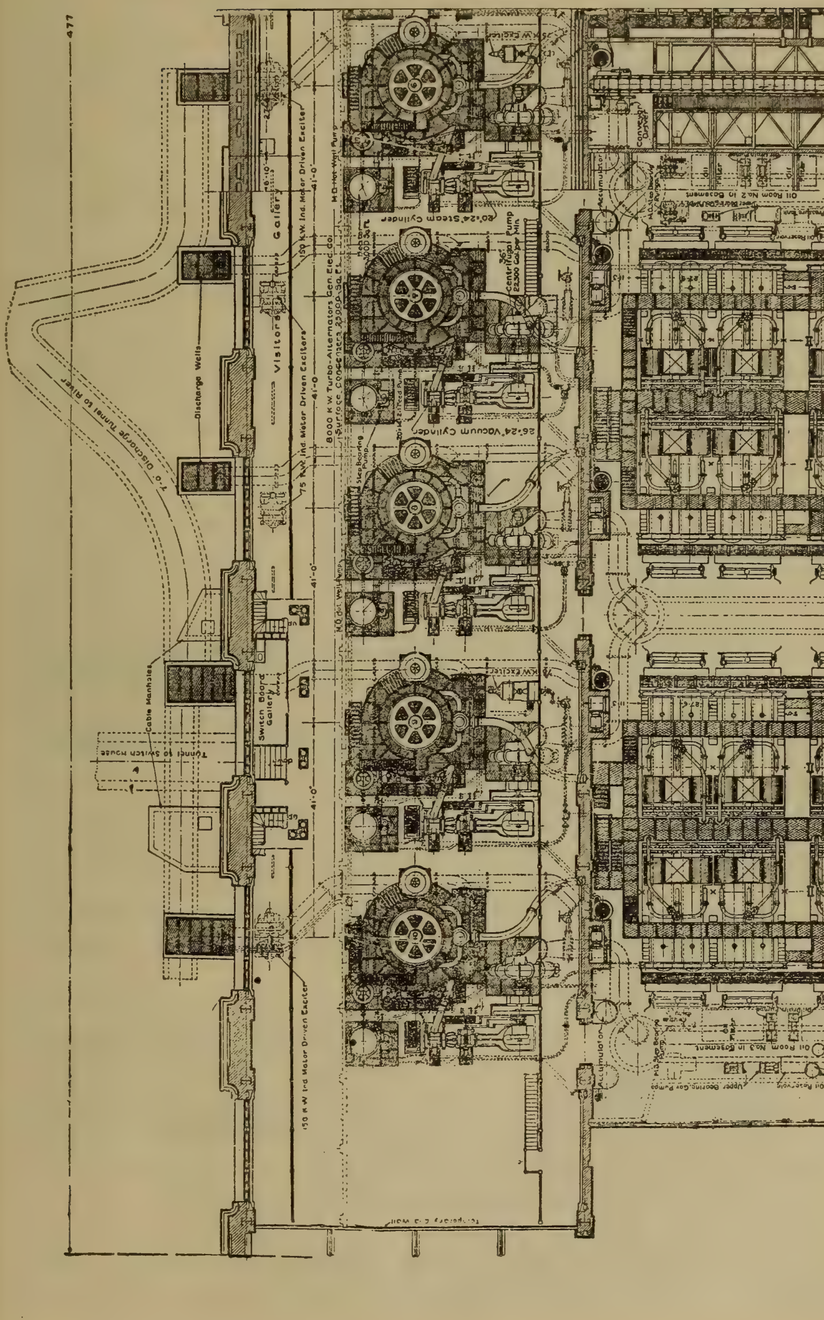


FIG. 595. General Plan of Units 6 to 10, Fisk Street Station.

consisting of sixteen 500-horse-power B. & W. boilers arranged in batteries of eight and equipped with B. & W. chain grates. The settings are installed back to back, as illustrated in Fig. 595. Each boiler has two 42-inch steam drums, approximately 5000 square feet of heating surface, and about 1000 square feet of superheating surface. Steam is generated at a pressure of 200 pounds per square inch with superheat of 150° F. The ratio of water-heating surface to grate surface is approximately 55 to 1, and the ratio of the total heating surface to grate surface is about 66 to 1. The grates are driven by Stokes motors with Krehbiel oscillating engines held in reserve. The boilers are supported by reënforced girders of the main building structure. A gallery is placed in front of the settings, 8 feet above the floor, to facilitate cleaning of tubes. Galleries are also placed between the batteries and on top of them. Spaces of 5 feet are provided between the sides and rears of the batteries, and 18 feet 8 inches in front. The furnaces are similar to the one illustrated in Fig. 97. The outside of the setting is finished with red pressed brick. Each drum is fitted with a $4\frac{1}{2}$-inch pop safety valve. The superheater is also fitted with a pop safety valve. The blow-off main is 4 inches in diameter and discharges into the river. There are four blow-off connections to each boiler, each being provided with a blow-off cock and an angle valve; three of the connections are fitted to the mud drum and the other to the superheater drain.

Chimneys. — One stack is provided for each section of 16 boilers. The shaft is supported by the steel work of the boiler setting, as shown in Fig. 596, an arrangement which commends itself where space is limited and real estate values are high. The stacks for all units are 259 feet 6 inches in height above the grate surface, and are 18 feet in internal diameter. The lining is of radial fire brick and varies from 4 inches to 13 inches in thickness. The steel sections are 5 feet high and vary in thickness from $\frac{3}{8}$ inch to $\frac{1}{2}$ inch. There are two flues, one 32 feet long and the other 63 feet, which enter the stack on opposite sides.

Turbines. — The prime movers are vertical five-stage Curtis turbines with base condenser and are rated at 12,000 kilowatts each. The normal speed is 750 r.p.m. The average steam consumption, including all auxiliaries, is approximately 15 pounds per kilowatt hour, corresponding to a coal consumption of 3 pounds per kilowatt hour (Illinois screenings, 10,400 B.t.u. per pound). Special tests have shown as low as 12.8 pounds per kilowatt hour, initial pressure 200 pounds gauge, 150 degrees superheat, absolute back pressure $\frac{1}{2}$ inch of mercury. Each pair of units has a pair of duplicate pumps, an accumulator, and a storage tank for supplying oil, the step-bearing pressure being maintained at

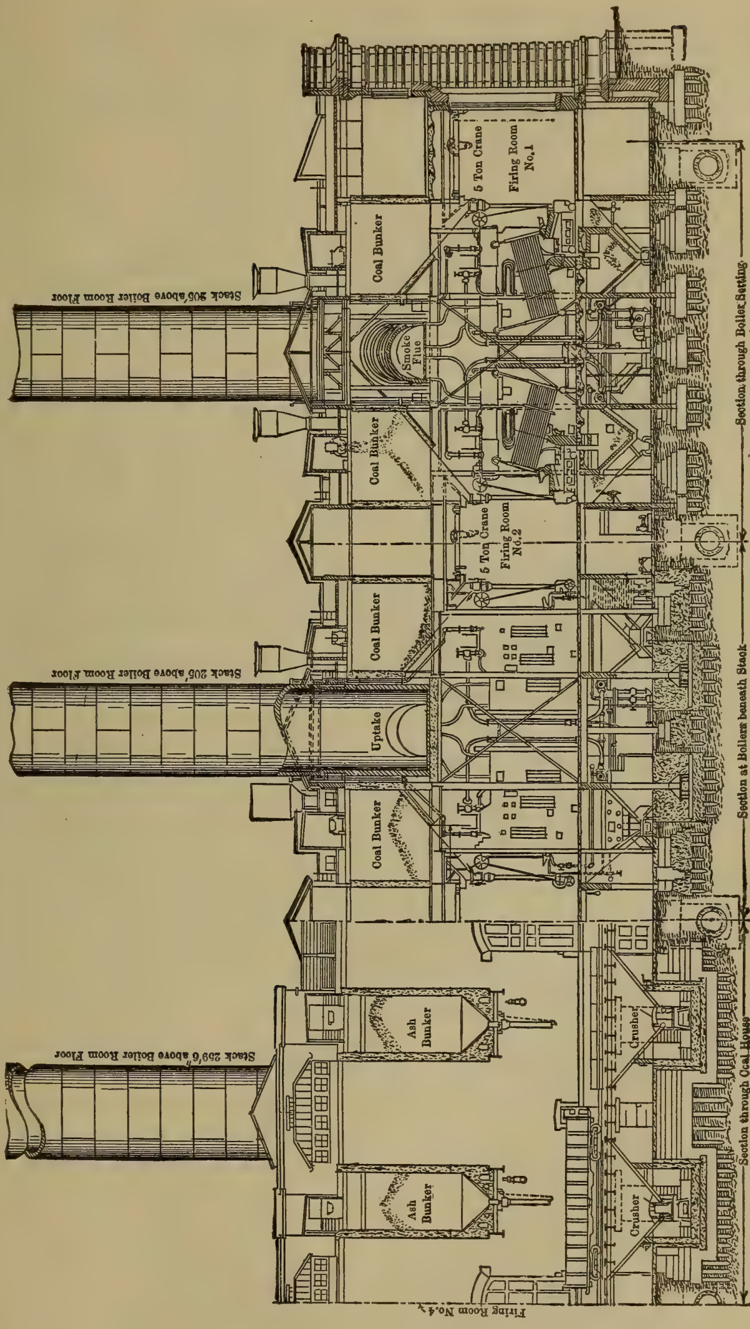


FIG. 596. Section through Boiler Room.

750 pounds per square inch. When the accumulator falls below a certain point a motor-driven pump is automatically started.

Generators. — The generators are 9000-volt, 25-cycle, three-phase General Electric alternators mounted over the vertical shaft as illustrated in Fig. 253. Exciting current is furnished by

Two 50-kilowatt motor-driven generators.
Two 75-kilowatt motor-driven generators.
Two 150-kilowatt motor-driven generators.
Two 75-kilowatt steam-driven generators.
Two 150-kilowatt steam-driven generators.

Part are held in reserve, though no particular units are maintained for the purpose. The high-tension cables lead from the generator through an underground tunnel to the switch house, located about 50 feet west of the main building. The oil switches, wattmeters, and other instruments are located on the first floor, while the bus-bars and other high-tension connections are in the basement. The station switchboard or operating gallery in the main building is equipped with only such devices as are necessary for the control of the machines, all other instruments being located in the switch house. From the switch house the high-tension current is conducted through oil switches to the various substations, where it is converted to direct current by rotary converters, or transformed from 25 to 60 cycles by motor generator sets.

The Twenty-second Street substation is located at the north end of the property (Fig. 593). In this substation are installed two motor generator sets and one rotary converter, the latter supplying direct current to the neighboring district and to the main station.

Boiler and Turbine Piping. — Immediately below each boiler section is an apartment called the "header room," where the steam pipes from the various boilers join the main header, which increases in size from 6 inches at the most remote boiler to 10 inches at the middle boiler, and finally to 14 inches where it leaves the nearest boiler and passes to the turbines. The pipes are of wrought iron, with welded flanges, and are packed with copper gaskets. The feeder from each boiler is 6 inches in diameter. An angle stop valve and a check valve are placed at the boiler nozzle and gate valves at the header. A motor-operated throttle valve and strainer are placed at the turbine, and the hydraulically operated valves are controlled from the turbine room or the header room. The main header is not anchored at any point, the entire weight being carried by roller supports. The only drain in the header is a $1\frac{1}{2}$-inch bleeder on the boiler side of the hydraulically operated valves. The bleeder is connected to a trap which discharges into either boiler or

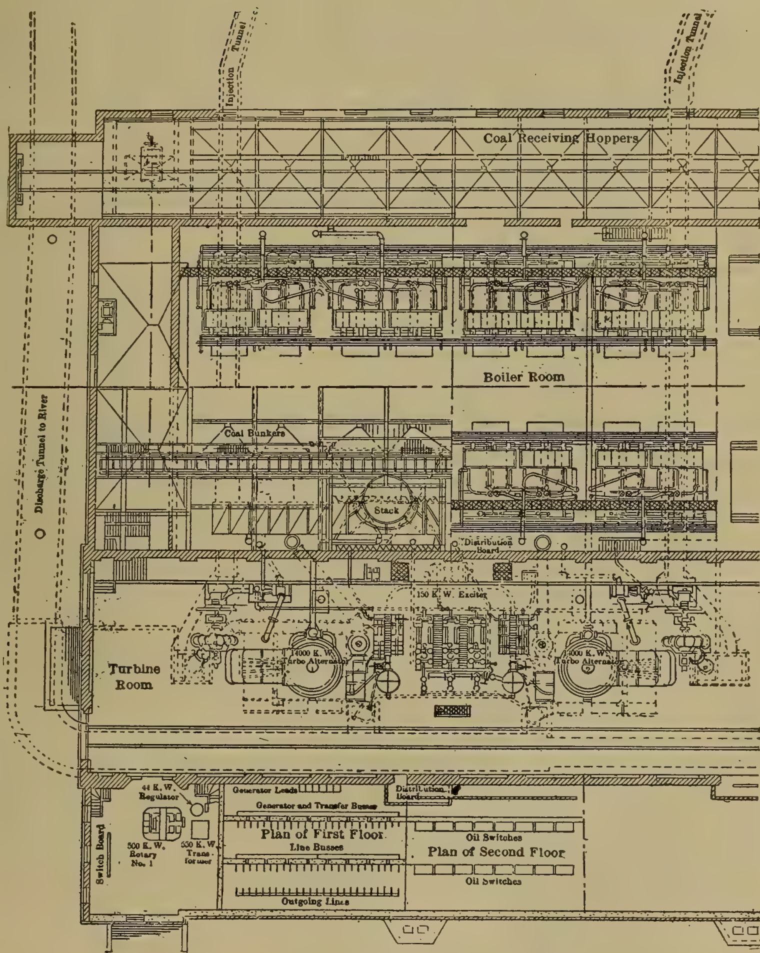


FIG. 597. General Plan of Quarry Street Station, Units 1 and 2.

superheater blow-off main. All branches or feeders are drained and discharged into the superheater blow-off.

Condensers and Auxiliaries. — Each unit has its own condensing apparatus, feed-water heater, hot well, and feed pumps. The condensers are of the Worthington "base" type with 25,000 square feet of cooling surface each, composed of 5900–6000 1-inch tubes 16 feet long. Cooling water is taken from the east slip through concrete tunnels and is discharged from the condenser into similar tunnels which empty into the west slip. (See Fig. 593.)

The dry vacuum pumps are of the rotative type, with cylinders 26 × 24, r.p.m. 100–120, and are driven by a 150-horse-power Corliss engine.

The circulating pumps are of the volute single-stage centrifugal type and are mounted on an extension of the main shaft of the engine driving the dry vacuum pump. They are rated at 22,500 gallons per minute each.

The hot-well pumps are of the two-stage centrifugal type, driven by 20-horse-power direct-current motors.

The feed pumps are of the Dean vertical single-cylinder pattern and are installed in duplicate for each unit. The steam cylinders are 24 inches in diameter, water cylinders 14 inches in diameter, and common stroke 24 inches. Feed water is pumped from the hot well at a temperature of about 100° F. and is forced through closed heaters having 3000 square feet of heating surface, and its temperature is raised to 180 degrees. The heater receives the steam exhausted from the steam-driven auxiliaries.

From the pumps this water is forced through a 5-inch feed main to the different boilers in the section. The branches from main to boiler drum are 3 inches in diameter and fitted with an angle stop valve, a regrinding check and a gate regulating valve. A combination stop and check valve is placed at the drum. There is a 5-inch auxiliary main which supplies cold water to the boiler in case the main header is shut down.

Miscellaneous. — The work is divided into eight-hour shifts. The list of operating men per unit is as follows:

In turbine room, including janitor work	1.0
In oil room	1.0
Attending water	0.5
Fireman	1.0
Fireman's helper	0.5
Conveyor men	1.0
Turbine switchboard gallery	0.3
Exciter tenders	0.2
Switchhouse attendants	0.2
	5.7

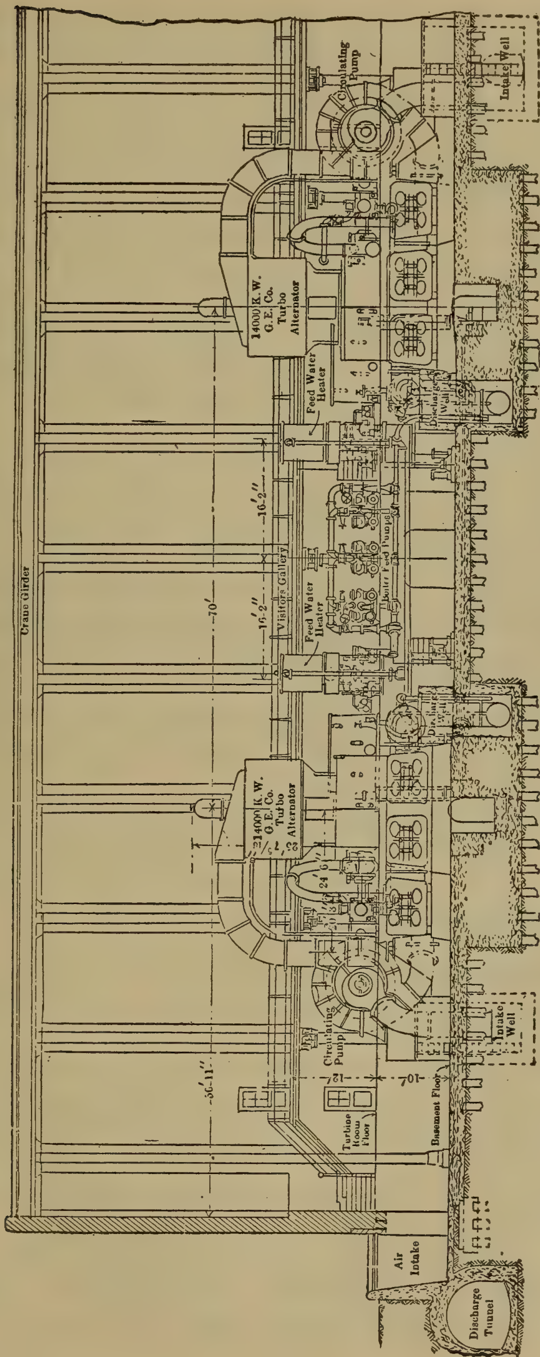


FIG. 598. Side Elevation, Quarry Street Station, Units 1 and 2.

Drains and drips from the auxiliaries empty into sumps from which they are discharged by Yeoman's bilge pumps into the discharge tunnel.

A steam-driven house pump is located in the basement.

The fire-protection system includes a 220-horse-power motor-driven spherical pump located in the basement and a connection for a fire tug.

Dining room, reading room, shower and tub baths, and sleeping rooms for emergencies are provided for the employees.

Quarry Street Station. — Figs. 597 to 599 give general views of the Quarry Street Station, which is located directly across the river from the Fisk Street Station. The two stations are distinct; a breakdown in one would not affect the other; nevertheless, they are operated together. That is, there is one chief engineer for the two, and the combined station force of 500 men can be shifted from one to the other as needed.

The general layout of the Quarry Street Station differs from that of the Fisk Street Station on account of real-estate limitations. The boilers are in two parallel rows instead of the equipment for each unit extending at right angles to the turbine room as at Fisk.

The station contains six 14,000-kilowatt Curtis turbo-alternators, the steam for each unit being supplied by eight 500-horse-power B. & W. boilers arranged as shown in Fig. 597. A 5000-kilowatt motor-generator set is also included in the equipment. Steam is generated at 225 pounds gauge pressure and 150–175 degrees superheat. The settings for the first two units are similar to that illustrated in Fig. 99, and the other similar to the one illustrated in Fig. 100.

A novel system of ventilation enables the generators to be operated continuously at full load. As will be seen from Fig. 598 air ducts lead from an outside intake to the top of each unit, the revolving portion of the generator being designed to draw in a continuous supply of air and discharge it through openings in the turbine casing.

Everything is in duplicate, so that the chance of breakdown is remote. Each unit is equipped with a volute centrifugal pump driven by a 125-horse-power Corliss engine. The water of condensation is removed from the condensers by hot-well pumps driven by Kerr steam turbines, one for each unit. For each two units of turbines and boilers three boiler feed pumps are provided, two centrifugal and one reciprocating, all of them located between the turbines. There are also four step-bearing oil pumps, two oil accumulators, dry-air pumps, oil filters, etc. All are in plain view of the turbine-room operating force.

For the six units there are five 150-kilowatt exciters, three driven by horizontal Curtis steam turbines and two by induction motors. In addition there is an excitation storage battery of 70 cells in the base-

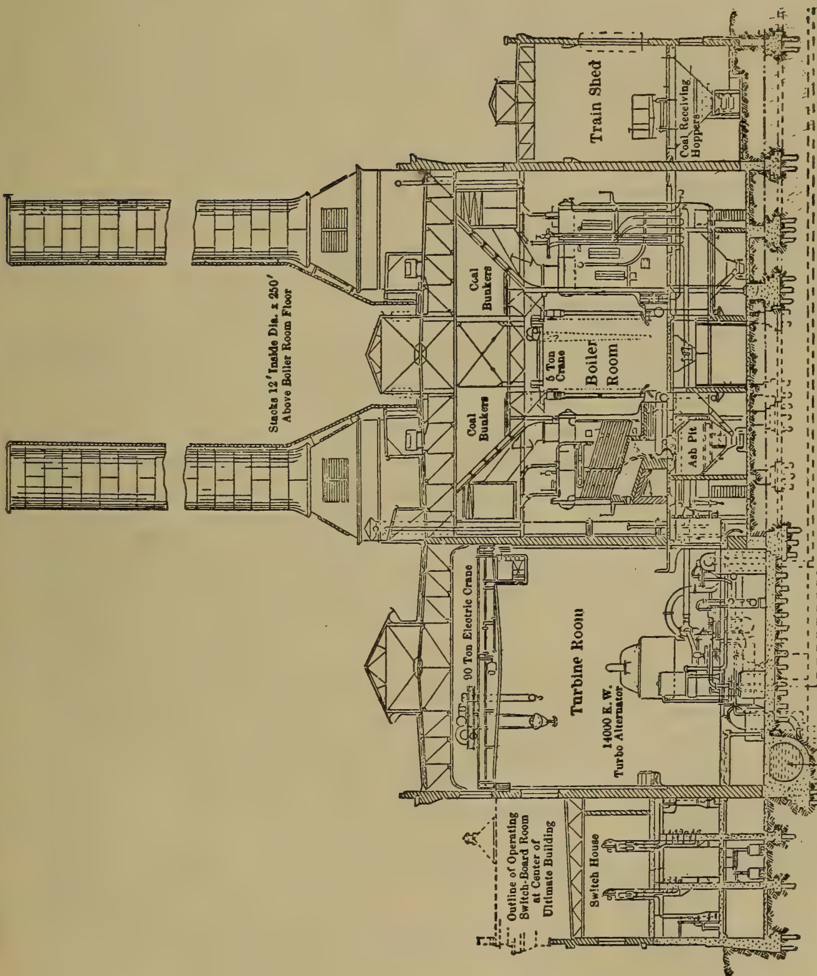


FIG. 599. Sectional Elevation, Quarry Street Station.

ment. Furthermore, in an emergency, 500-kilowatt rotary converters of the substation could be used for excitation.

Northwest Stations. — Two new stations of 120,000-kilowatt rated capacity each are to be installed on the north branch of the Chicago River near Roscoe Street and California Avenue. Each station is to be equipped with six 20,000-kilowatt Curtis turbo-generators, 2300-volt, 25-cycle, three-phase, 750-r.p.m., similar to those installed at Quarry Street. The first two units for one station are now in operation. Each unit is supplied with steam at 250 pounds gauge pressure and 150 degrees superheat from ten 500-horse-power B. & W. boilers. The boiler settings are similar to the one illustrated in Fig. 100. The general layout is the same as at Fisk, the boiler lanes extending at right angles to the turbine room. There is one chimney 17 feet inside diameter and 250 feet in height for every ten boilers.

The present capacity (November, 1912) of the Commonwealth Edison Company is about 280,000 kilowatts, divided as follows:

	Units.	Present Capacity.	Ultimate Capacity.
Fisk	10–12,000	120,000	170,000*
Quarry	6–14,000	84,000	84,000
Northwest No. 1	6–20,000	40,000	120,000
Northwest No. 2	6–20,000	....	120,000
Miscellaneous Plants	....	36,000	36,000
		280,000	530,000

* 2-25,000-kilowatt Parsons turbines are now being added to the present equipment.

COMPARATIVE BOILER ROOM AND ENGINE ROOM AREAS.

	Fisk.	Quarry.	Northwest.
Boiler room, sq. ft. per kw	0.51	....	0.44
Engine room, sq. ft. per kw	0.24	0.18	0.15
Total area, sq. ft. per kw	0.75	....	0.59

For a detailed description of the Northwest Station, consult Practical Engineer, U. S., Jan. 15, 1913, p. 109.

CHAPTER XXI.

A TYPICAL MODERN ISOLATED STATION.*

Bleeder Turbines and Condenser System.

THE new power plant of the W. H. McElwain Company at Manchester, N. H., is an excellent example of current practice in generation of power by steam for industrial purposes.

General Arrangement. — General arrangement of the boiler and engine rooms is shown in plan in Fig. 601. At the present time there have been installed three 300-horse-power water-tube boilers and one 1000-kilowatt turbo-generator outfit. The boiler room contains sufficient space for a fourth 300-horse-power unit, as indicated by dotted lines. The completed plant will include duplicates of the two batteries shown, making a total of 2400 horse power. The future boilers will face those already installed, the building being extended for this purpose, and the firing space shown will be common to both sections.

The chimney, which is 176 feet in height, with a flue 9 feet in diameter, is designed with reference to the final capacity of the plant. In the engine room, at the right, is shown space for two additional generating units, which provide for an ultimate capacity of 3000 kilowatts. Sectional elevations, showing the boilers, turbines, and the various auxiliary equipment and their connections, are illustrated in Figs. 600, 602, and 603.

Boilers. — Present equipment consists of three Babcock and Wilcox water-tube boilers, each containing 2972 square feet of heating surface and about 50 square feet of grate surface. The heating surface is made up of two steam drums, tubes, and mud drum, and a superheater of the form shown in Fig. 602.

Each boiler contains 144 4-inch tubes, 18 feet in length, made up in 12 sections of 12 tubes each, and 2 steam drums, 3 feet in diameter by 20 feet 4 inches in length. The superheaters each contain approximately 372 square feet of surface, which is $12\frac{1}{2}$ per cent of the heating surface of the boiler, and are designed to give 100 degrees superheat when the boilers are operated at their normal rating of 300 horse power. The proportions of all parts are designed for a working pressure of 160 pounds per square inch and the safety valves are set at that point.

* From the Practical Engineer, Chicago, July 1, 1912.

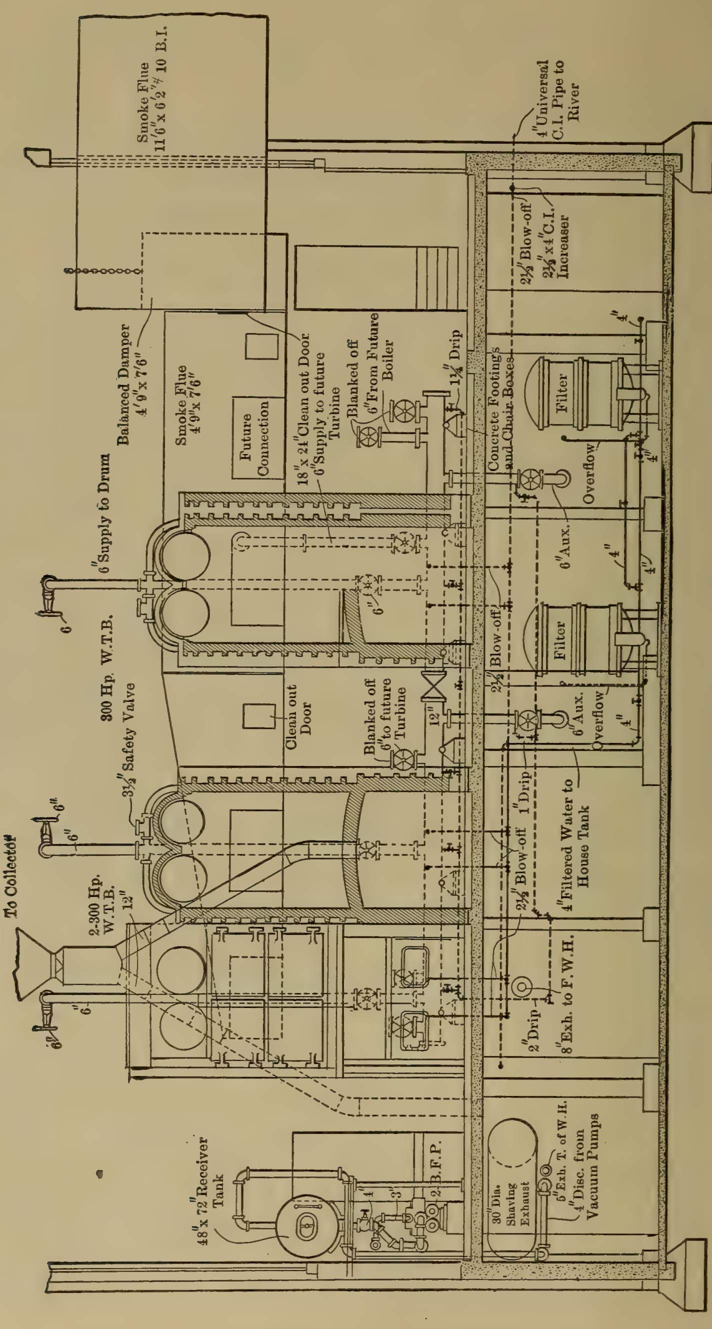


FIG. 600. Sectional Elevation through Boiler Room and Basement.

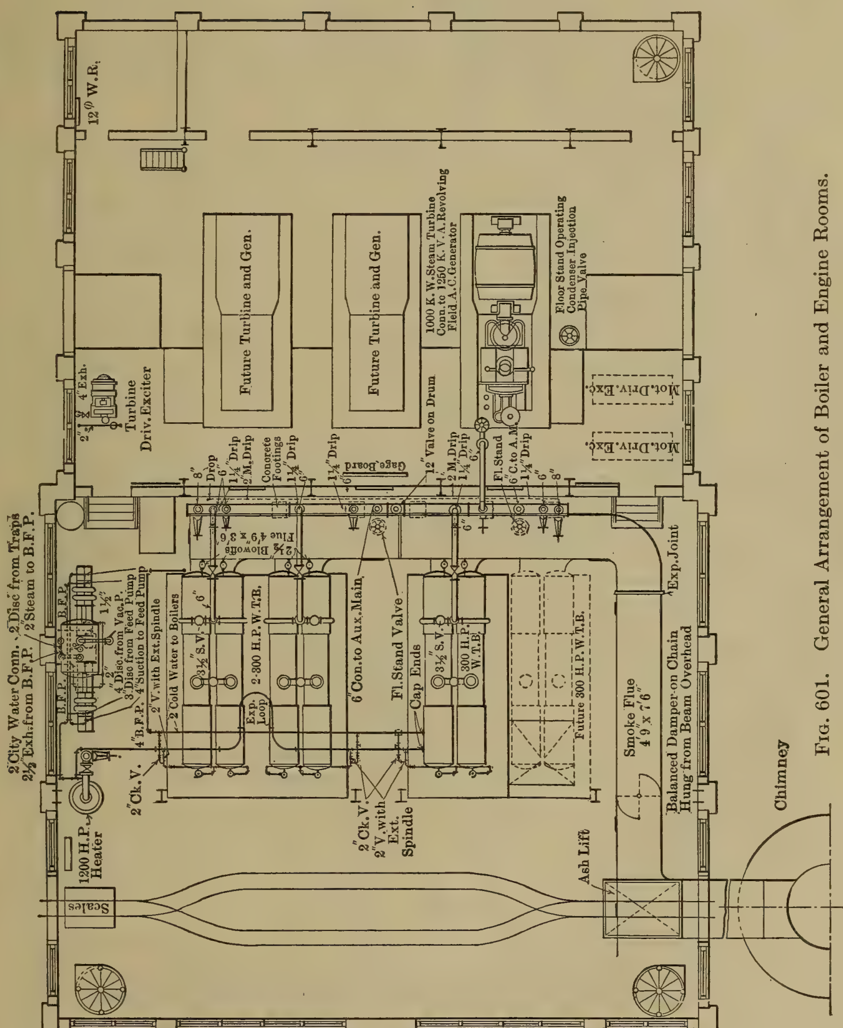


FIG. 601. General Arrangement of Boiler and Engine Rooms.

Each boiler is provided with a water column fitted with high and low water alarm, try cocks, and gauge glass with special device for shutting off in case of breakage. Also $3\frac{1}{2}$-inch lock pop-safety valve, and 12-inch steam gauge reading to 300 pounds pressure. The feed pipes are 2 inches in diameter, provided with both check and gate valves, the latter having special extension handles. The blow-off connections are of $2\frac{1}{2}$-inch extra heavy pipe, and are each provided with two blow-off valves of special design.

Boiler settings are of hard-burned brick, laid in cement mortar, consisting of 1 part cement to 3 of sand, up to the level of the grates, and in lime mortar above that point. All parts of the furnaces and setting exposed to the fire are lined with firebrick laid in fire clay. The furnaces are of the "Dutch oven" type as shown in Fig. 602.

Smoke Connections. — Location of the main smoke flue is best shown in Fig. 601. It is 4 feet 9 inches by 7 feet 6 inches in size and constructed of No. 10 black iron. It is stiffened with angle-iron braces and supported from the roof. The uptake from each boiler is provided with an adjusting damper for hand manipulation from the floor level.

A balanced damper is located in the main flue at the point indicated, and operated by an automatic regulator of the hydraulic type. An interesting detail in connection with this work is the method of attaching the covering to the lower side of the flue so that it will not sag or peel off. This consists of cross pieces of 1-inch tee-bars placed 24 inches apart and riveted to the flue. The projecting flanges of these bars are drilled at frequent intervals and wires strung through, to which the covering is attached.

Handling of Fuel and Ash. — Coal is brought to the fire room by cars running on a special track as shown in Fig. 601. This track passes over platform scales just inside the building, where each load may be weighed as it is brought in. The track is double within the fire room so that the cars may pass, and also to furnish storage space for both coal and ash cars when so desired.

The arrangement for the removal of ash is best shown in Figs. 601 and 602. A dumping chute is provided in the bottom of each ashpit and at such an elevation that a car may be run underneath it as indicated. When filled, they are pushed to the ash lift (see Fig. 601) where they are raised to the boiler-room level and run out on the coal track for disposal. Combustible waste from the factory is brought through a 36-inch pipe to a collector placed in the upper part of the boiler room, as shown in Fig. 602, and fed into the furnaces as there indicated.

Turbine and Generator. — The turbo-generator unit is one of the Westinghouse make, of 1000-kilowatt capacity, and equipped with an automatic bleeder connection and constant-pressure valve. It is 6 feet

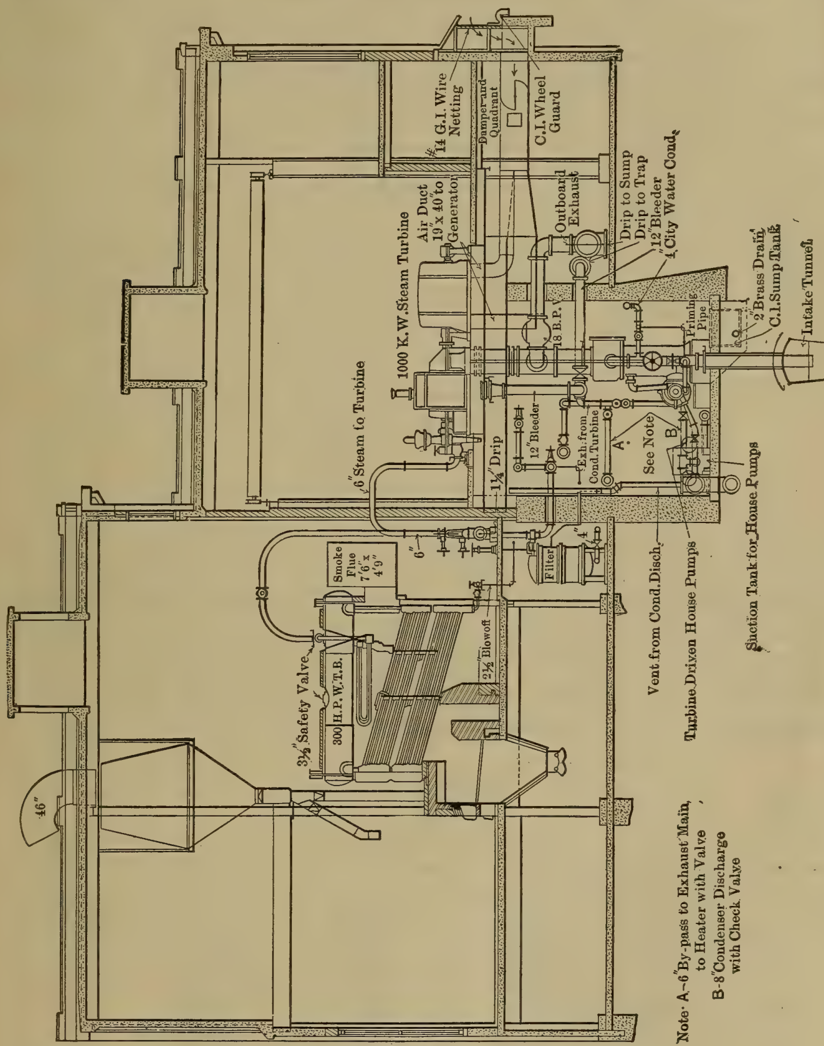


FIG. 602. Sectional Elevation through Boiler and Engine Room, showing Steam Piping and Condenser Connections.

6 inches in width by 24 feet 8 inches in length and weighs approximately 79,000 pounds. It is of the regular Westinghouse-Parsons type, the most interesting feature being the bleeder attachment which adapts it for use in combined power and heating plants. An important requirement for the economical operation of the ordinary steam turbine

is the maintenance of a high vacuum at the exhaust end, which, of course, prevents the utilization of exhaust steam for heating purposes.

The capacity of the turbine under different conditions is as follows: With a throttle pressure of 150 pounds per square inch (gauge), a vacuum of 28 inches, 100 degrees superheat, and a speed of 3600 r.p.m., the normal capacity when condensing is 1500 b.h.p. and the maximum 2250 b.h.p. When running non-condensing with a back pressure not exceeding that of the atmosphere, the maximum capacity is 1500 b.h.p.

It is interesting to note the probable steam economy of a turbine of this type when operating under varying loads, as expressed in the guarantee placed upon this machine, which is as follows: When operating under the above conditions, in connection with the generator attached, the steam consumption per hour, including all leakage and loss with the turbine, shall not exceed the quantities given below:

Load, Per Cent.	Power Factor, Per Cent.	Kilowatts.	Pounds Steam per Kilowatt-Hour.
150	80	1500	18.8
125	80	1250	18.3
100	80	1000	17.9
75	80	750	18.8
50	80	500	20.7

When operating under the same general conditions, with 3 pounds gauge pressure at the bleeder connection, the steam consumption per hour shall not exceed the following at the loads indicated, when withdrawing the following amounts of steam through the bleeder connection:

Load, Per Cent.	Kilowatts.	Pounds of Steam		Steam to Condenser.	
		To Throttle.	To Bleeder.	Total.	Kilowatts.
150	1500	38,000	18,600	19,400	12.9
		31,000	10,000	21,000	14.0
125	1265	38,000	22,000	16,000	12.7
		36,300	20,000	16,300	12.9
		29,200	10,000	19,200	15.2
100	1000	37,200	30,000	7,200	7.2
		30,000	20,000	10,000	10.0
		24,400	10,000	14,400	14.4
75	716	30,500	30,000	500	0.7
		25,600	20,000	5,600	7.8
		20,200	10,000	10,200	14.2
50	469	21,700	21,700	0	0.0
		20,600	20,000	600	1.3
		16,000	10,000	6,000	12.8

Generator. — The generator is of the revolving-field type with inclosed frame, generating a 3-phase, 60-cycle, alternating current of 600 volts. The efficiency rating, with a power factor of 100 per cent, is as follows:

Load, Per Cent.	Efficiency, Per Cent.	Load, Per Cent.	Efficiency, Per Cent.
50	90.10	125	95.50
75	93.00	150	95.75
100	94.50		

Temperature rise based on its normal rating and a power factor of 80 per cent, for periods of different length and for various loads, is given below:

Load, Per Cent.	Length of Run, Hours.	Temperature Rise, Armature.	Degs. F., Field.
100	24	72	72
125	24	90	90
150	1	108	108

The maximum conditions of continuous operation with a power factor of 80 per cent and for a room temperature of 77 degrees F. are as follows: Output, 1250 kilowatts (25 per cent overload). Rise in temperature:

Armature, 90 degrees F.
Field, 90 degrees F.

Maximum temperature to which insulation can be subjected without injury:

Armature, 194 degrees F.
Field, 302 degrees F.

There are two exciters provided, one being turbine driven and having a normal capacity of 25 kilowatts; the other motor driven, with a capacity of 40 kilowatts. The turbine is of the Westinghouse make, horizontal type, with a normal capacity of 38 b.h.p. at a speed of 3500 r.p.m. when running non-condensing, and a continuous overload capacity of 25 per cent. The steam requirements for this machine as regards temperature and pressure are the same as for the main turbine.

The exciter is a direct-current machine with shunt winding, generating a current of 125 volts at full load.

Condensing Apparatus. — In connection with the main turbine a Westinghouse-LeBlanc jet condenser is used, and is shown in elevation in Figs. 602 and 603. This is designed to operate under a normal lift

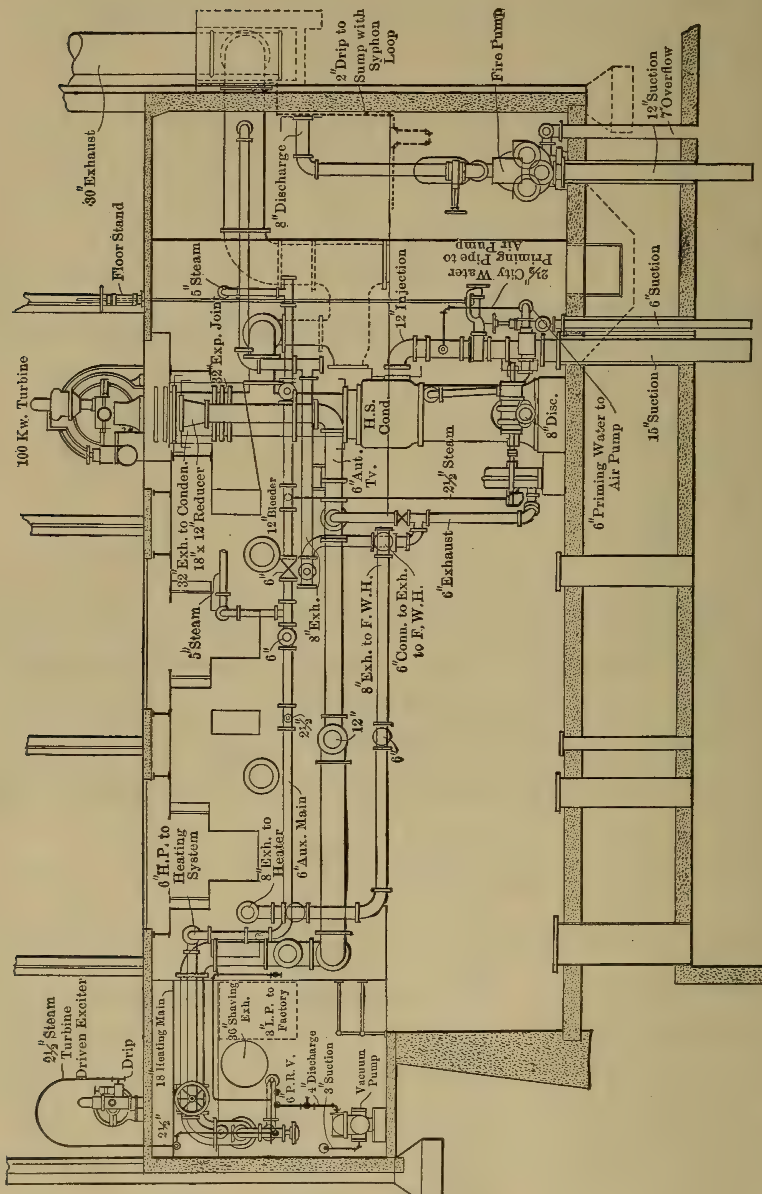


Fig. 603. Longitudinal Section of Engine Room Basement, showing Condenser Connections and Piping to Heating System.

of 18 feet and takes its water supply from the intake tunnel as shown. When using injection water at a temperature of 70 degrees F. the following results are guaranteed, with a water consumption not exceeding 724,000 pounds per hour:

Steam Condensed per Hour, Pounds.	Vacuum Maintained, Inches (Barometer, 30 Ins.)	Steam Condensed per Hour, Pounds.	Vacuum Maintained, Inches (Barometer, 30 Ins.)
10,350	28.65	19,950	28.00
14,100	28.44	22,900	27.80
18,000	28.17	30,000	27.11

The vacuum air pump is of the turbine type and is mounted upon the same shaft with the centrifugal ejector pump, both being driven by a steam turbine of 41 b.h.p. running at 1500 r.p.m. under an atmospheric exhaust pressure. This piece of apparatus is shown at the base of the condenser in Figs. 602 and 603.

High-pressure Piping System. — This includes all high-pressure piping in the boiler and engine rooms for the supply of turbines, pumps, etc., and for the supplementary supply to the heating system as may be needed. Pipe used for this purpose is full weight, wrought iron being used for sizes below 6 inches and open-hearth steel for larger sizes. The main drum at the rear of the boilers is of gun metal with nozzles cast in place. Expansion is provided for, so far as possible, by the use of sweep pipe bends and fittings of the long-turn pattern, all 2½-inch and larger fittings being of this design with flange joints. The high-pressure connections are shown in Figs. 601, 602, and 604. Starting at the boilers (Fig. 602), 6-inch leads are carried to a 12-inch drum supported on low piers and rolls at the rear of the boilers. From here a 6-inch branch leads to the main turbine, and two branches of the same size to a 6-inch auxiliary main, running beneath the engine-room floor, near to, and parallel with, the boiler-room wall. From this auxiliary main are taken the supplies to the various minor turbines and pumps, and also the branches leading to the low and intermediate-pressure system through reducing valves. The main drum is divided into two sections by means of a valve at the center, and each of these sections is connected with the auxiliary drum as shown in Figs. 600 and 604. The supplies to the various pumps are easily traced from Fig. 604, also the connections with the 18-inch heating main and the intermediate-pressure line, leading to the factory through the tunnel leaving the building as indicated in the upper right-hand corner of the drawing.

Exhaust System. — All low and intermediate pressure piping is full weight, sizes up to, and including, 12-inch being of wrought iron, while

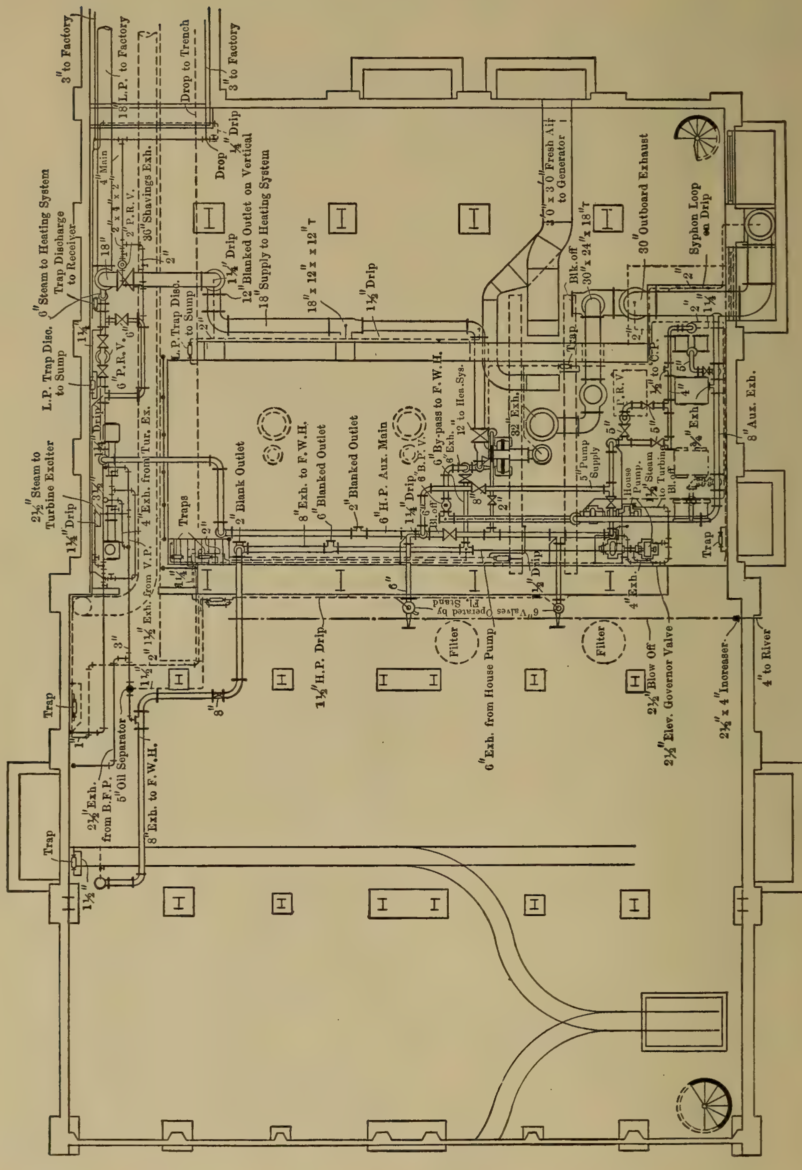


FIG. 604. Plan of Steam and Exhaust Piping.

open-hearth steel is employed for the larger sizes. Standard-weight fittings are used for this work, those 6 inches and over being of the long-turn pattern. Flange joints are provided on all piping $2\frac{1}{2}$ inches and larger in diameter, the same as for high-pressure work. The exhaust piping is most clearly shown in Figs. 602, 603, and 604. Referring to Fig. 604 the 18-inch exhaust from the main turbine is shown as leading through a back-pressure valve into a 30-inch outboard line designed for the completed plant. This is clearly shown in elevation in Fig. 603. An 8-inch auxiliary exhaust connecting with the various pumps is shown in Fig. 603, parallel with, and below, the auxiliary high-pressure main already described. Steam from this enters the heating system through an oil separator. The 12-inch bleeder connection from the turbine leads to the 18-inch heating main and is shown in the same drawing, although more clearly in Fig. 604.

Drainage.—The blow-off main from the boilers is carried directly to the river through a 4-inch cast-iron pipe. Drips from high-pressure piping are trapped to the main receiving tank and pumped back to the boilers. Exhaust drips, and all condensation containing oil, are trapped to a cast-iron sump tank located in the condenser pit, and, together with other drainage, are discharged by means of a water ejector.

Water Supply, Feed Piping, etc. — Water for condensing and fire purposes is brought from the river through a cement conduit, a section of this, together with the 15-inch suction to the condenser, being shown in Figs. 602 and 603. The discharge from the condenser pump is into an 18-inch pipe leading to the river and shown in section in Fig. 602. Water pressure for fire protection is furnished by an 18 by 10 by 12-inch Underwriters' fire pump of 1000 gallons capacity, placed in the condenser pit; this is shown in elevation in Fig. 603 and in plan in Fig. 604 and takes its supply from the intake tunnel as there shown.

The house tank and boilers have two sources of supply, one directly from the city mains and the other from the intake tunnel. There is also a tank arrangement whereby water may be drawn from the discharge pipe of the condenser pump.

These various lines are shown in Fig. 605. A 6-inch connection from the city main enters as shown at the upper part of the drawing, toward the left, and, after passing through a meter, branches are carried to the house tank, the receiving tank, the boilers, and to the priming pipes of the condenser and vacuum pumps.

The second source of supply, that from the intake tunnel, requires the use of two turbine-driven house pumps of the one-stage turbine type, located in the condenser pit as shown in Fig. 605. These pumps each have a capacity of 200 gallons per minute against a head

of 150 feet, and discharge into a line of piping having branches connecting with the house tank, receiving tank, and boiler feed pipe. A filtering equipment is also provided, as shown in Fig. 605, and so connected that the water from this source may be purified if desired.

Boiler Feed. — Feed lines connecting with the boilers are shown in Fig. 601. One of these supplies water either directly by city pressure

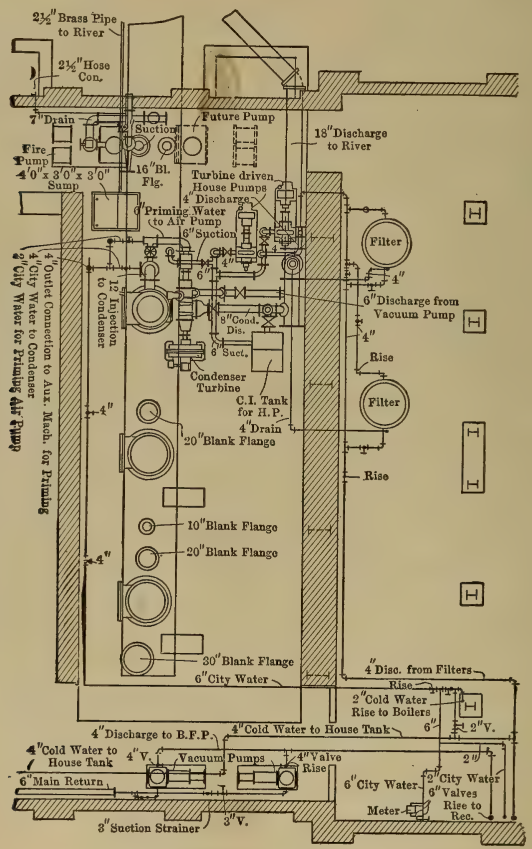


FIG. 605. Plan of Condenser Piping.

or from the turbine house pumps. The other supply is from a pair of boiler feed pumps connecting with a receiving tank located in the boiler room as shown. The feed pumps, two in number, are of the duplex, outside packed, pot-valve type, 8 by 5 by 10 inches in size. The tank is 4 feet in diameter by 6 feet in length, of $\frac{3}{8}$-inch iron plate, and is connected with both city pressure and the house pumps. Under ordinary working conditions the feed supply is first discharged into the

tank and then pumped to the boilers through a heater of 1000-horsepower capacity located as shown in the drawing.

Heating System. — Factory buildings are heated by direct radiation in the form of coils and cast-iron radiators as best suited to local conditions. The Webster system of circulation is employed, a pair of 6 by 10 by 12-inch single-piston vacuum pumps being connected with the main return as shown in Fig. 605. These discharge into the receiving tank in the boiler room, and the condensation is pumped back to the boilers with the fresh feed.

Steam supply for the radiation has already been mentioned, coming principally through the bleeder connection from the main turbine, supplemented, when necessary, by live steam through a reducing valve.

Insulation. — In general, tanks, heaters, etc., are covered with 85 per cent magnesia blocks, finished with a plastic coat of the same material, the total thickness of the covering, when finished, being 2 inches. In addition to this, tanks and heater are provided with a covering of 7-ounce canvas. The insulation on that portion of the smoke pipe which comes outside of the building is protected by a covering of heavy sheet iron. Steam piping, both high and low pressure, is insulated with 85 per cent magnesia sectional covering. All cold-water piping, with the exception of the connections to the condenser, are covered with wool felt, having a lining of tarred paper. Pipe covering of all kinds is finished with a heavy canvas jacket and painted.

APPENDIX A.

INSTRUCTIONS REGARDING TESTS IN GENERAL.*

A.S.M.E. CODE OF 1912.

1. OBJECT.

Ascertain the specific object of the test, and keep this in view not only in the work of preparation, but also during the progress of the test, and do not let it be obscured by devoting too close attention to matters of minor importance. Whatever the object of the test may be, accuracy and reliability must underlie the work from beginning to end.

If questions of fulfillment of contract are involved, there should be a clear understanding between all the parties, preferably in writing, as to the operating conditions which should obtain during the trial, and as to the methods of testing to be followed, unless these are already expressed in the contract itself.

Among the many objects of performance tests, the following may be noted:

Determination of capacity and efficiency, and how these compare with standard or guaranteed results.

Comparison of different conditions or methods of operation.

Determination of the cause of either inferior or superior results.

Comparison of different kinds of fuel.

Determination of the effect of changes of design or proportion upon capacity or efficiency, etc.

2. PREPARATIONS.

(A) Dimensions.

Measure the dimensions of the principal parts of the apparatus to be tested, so far as they bear on the objects in view, or determine these from correct working drawings. Notice the general features of the same, both exterior and interior, and make sketches, if needed, to show unusual points of design.

(B) Examination of Plant.

Make a thorough examination of the physical condition of all parts of the plant or apparatus which concern the object in view, and record the conditions found, together with any points in the matter of operation which bear thereon.

* Preliminary report of the Committee on Power Tests (Jour. A.S.M.E., Nov., 1912). Greatly abridged.

If the object of the test is to determine the highest efficiency or capacity obtainable, any physical defects, or defects of operation, tending to make the result unfavorable should first be remedied; all fouled parts being cleaned, and the whole put in first-class condition. If, on the other hand, the object is to ascertain the performance under existing conditions, no such preparation is either required or desired.

(*C*) *General Precautions against Leakage.*

In steam tests make sure that there is no leakage through blow-offs, drips, etc., or any steam or water connections of the plant or apparatus undergoing test, which would in any way affect the results. All such connections should be blanked off, or satisfactory assurance should be obtained that there is leakage neither out nor in. This is a most important matter, and no assurance should be considered satisfactory unless it is susceptible of absolute demonstration.

(*D*) *Apparatus and Instruments.*

Select the apparatus and instruments specified in the Code of Rules applying to the test in hand, locate and install the same, and complete the preparations for the work in view.

The arrangement and location of the testing appliances in every case must be left to the judgment and ingenuity of the engineer in charge, the details being largely dependent upon locality and surroundings. One guiding rule, however, should always be kept in view, viz., see that the apparatus and instruments are substantially reliable, and arrange them in such a way as to obtain correct data.

3. MISCELLANEOUS INSTRUCTIONS.

The person in charge of a test should have the aid of a sufficient number of assistants, so that he may be free to give special attention to any part of the work whenever and wherever it may be required. He should make sure that the instruments and testing apparatus continually give reliable indications, and that the readings are correctly recorded. He should also keep in view, at all points, the operation of the plant or part of the plant under test and see that the operating conditions determined on are maintained and that nothing occurs, either by accident or design, to vitiate the data. This last precaution is especially needed in guarantee tests.

Before a test is undertaken, it is important that the boiler, engine, or other apparatus concerned, shall have been in operation a sufficient length of time to attain working temperatures and proper operating conditions throughout, so that the results of the test may express the true working performance.

An exception should be noted where the object of the test is to obtain the working performance, including the effect of preliminary heating, in which case all the conditions should conform to those of regular service.

In preparation for a test to demonstrate maximum efficiency, it is desirable to run preliminary tests for the purpose of determining the most advantageous conditions.

4. OPERATING CONDITIONS.

In all tests in which the object is to determine the performance under conditions of maximum efficiency, or where it is desired to ascertain the effect of predetermined conditions of operation, all such conditions which have an appreciable effect upon the efficiency should be maintained as nearly uniform during the trial as the limitations of practical work will permit. In a stationary steam plant, for example, where maximum efficiency is the object in view, there should be uniformity in such matters as steam pressure, times of firing, quantity of coal supplied at each firing, thickness of fire, and in other firing operations; also in the rate of supplying the feed water, in the load on the engine or turbine, and in the operating conditions throughout. On the other hand, if the object of the test is to determine the performance under working conditions, no attempt at uniformity is either desired or required unless this uniformity corresponds to the regular practice, and when this is the object the usual working conditions should prevail throughout the trial.

5. RECORDS.

A log of the data should be entered in notebooks or on blank sheets suitably prepared in advance. This should be done in such manner that the test may be divided into hourly periods, or, if necessary, periods of less duration, and the leading data obtained for any one or more periods as desired, thereby showing the degree of uniformity obtained.

The readings of the various instruments and apparatus concerned in the test, other than those showing quantities of consumption (such as fuel, water, and gas), should be taken at intervals not exceeding half an hour and entered in the log. Whenever the indications fluctuate, the intervals should be reduced according to the extent of the fluctuation. In the case of smoke observations, for example, it is often necessary to take observations every minute, or still oftener, continuing these throughout the period covering the range of variations.

Make a memorandum of every event connected with the progress of a test, however unnecessary at the time it may appear. A record should be made of the exact time of every such occurrence and the

time of taking every weight and every observation. For the purpose of identification the signature of the observer and the date should be affixed to each log sheet or record.

In the simple matter of weighing coal by the barrow-load, or weighing water by the tank-full, which is required in many tests, a series of marks, or tallies, should never be trusted. The time each load is weighed or emptied should be recorded. The weighing of coal should not be delegated to unreliable assistants, and, whenever practicable, one or more men should be assigned solely to this work. The same may be said with regard to the weighing of feed water.

6. DATA REPRESENTED GRAPHICALLY.

If it is desired to show the uniformity of the data at a glance the whole log of the trial should be plotted on a chart, using horizontal distances to represent times of observation, and vertical distances on suitable scales to represent various data as recorded. Such a chart showing log of a boiler test is illustrated in Appendix No. 23.

It is instructive to plot the leading data on such a chart while the test is in progress.

7. REPORT.

The report of a test should present all the leading facts bearing on the design, dimensions, condition, and operation of the apparatus tested, and should include a description of any other apparatus and auxiliaries concerned, together with such sketches as may be needed for a clear understanding of all points under consideration. It should state clearly the object and character of the test, the methods followed, the conditions maintained, and the conclusions reached, closing with a tabular summary of the principal data and results.

APPENDIX B.

RULES FOR CONDUCTING EVAPORATIVE TESTS OF BOILERS.*

A.S.M.E. Code of 1912.

1. OBJECT AND PREPARATIONS.

Determine the object, take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions and make preparations for the test accordingly.

2. FUEL.

Determine the character of fuel to be used. For tests of maximum efficiency or capacity of the boiler to compare with other boilers, the coal should be of some kind which is commercially regarded as a standard for the locality where the test is made.

A coal selected for maximum efficiency and capacity tests should be the best of its class, and especially free from slagging and unusual clinker-forming impurities.

For guarantee and other tests with a specified coal containing not more than a certain amount of ash and moisture, the coal selected should not be higher in ash and in moisture than the stated amounts, because any increase is liable to reduce the efficiency and capacity more than the equivalent proportion of such increase.

The size of the coal, especially where it is of the anthracite class, should be determined by screening a suitable sample.

3. APPARATUS AND INSTRUMENTS.

The apparatus and instruments required for boiler tests are:

(*a*) Platform scales for weighing coal and ashes.
(*b*) Graduated scales attached to the water glasses.
(*c*) Tanks and platform scales for weighing water (or water meters calibrated in place).
(*d*) Pressure gages, thermometers, and draft gages.
(*e*) Calorimeters for determining the calorific value of fuel and the quality of steam.
(*f*) Furnace pyrometers.
(*g*) Gas analyzing apparatus.

* Preliminary report of the Committee for Power Tests. (Jour. A.S.M.E., Nov., 1912.) Greatly abridged.

Full directions regarding the use and calibration of the above mentioned appliances are given in the complete code.

4. OPERATING CONDITIONS.

Determine what the operating conditions and method of firing should be to conform to the object in view, and see that they prevail throughout the trial, as nearly as possible.

5. DURATION.

The duration of tests to determine the efficiency of a hand-fired boiler should be 10 hours of continuous running, or such time as may be required to burn a total of 250 pounds of coal per square foot of grate.

In the case of a boiler using a mechanical stoker, the duration, where practicable, should be at least 24 hours. If the stoker is of a type that permits the quantity and condition of the fuel bed at beginning and end of the test to be accurately estimated, the duration may be reduced to 10 hours, or such time as may be required to burn the above-noted total of 250 pounds per square foot.

6. STARTING AND STOPPING.

The conditions regarding the temperature of the furnace and boiler, the quantity and quality of the live coal and ash on the grates, the water level, and the steam pressure, should be as nearly as possible the same at the end as at the beginning of the test.

7. RECORDS.

The records of data should be obtained as pointed out in Appendix A. Half-hourly readings of the instruments are usually sufficient. If there are sudden and wide fluctuations, the readings in such cases should be taken every fifteen minutes, and in some instances oftener.

8. QUALITY OF STEAM.

If the boiler does not produce superheated steam the percentage of moisture in the steam should be determined by the use of a throttling or separating calorimeter. If the boiler has superheating surface, the temperature of the steam should be determined by the use of a thermometer inserted in a thermometer well.

9. SAMPLING AND DRYING COAL.

During the progress of the test the coal should be regularly sampled for the purpose of analysis and determination of moisture.

10. ASHES AND REFUSE.

The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed so far as possible in a dry state. If wet the amount of moisture should be ascertained and allowed for, a sample being taken and dried for this purpose. This sample may serve also for analysis and the determination of unburned carbon and fusing temperature.

11. CALORIFIC TESTS AND ANALYSES OF COAL.

The quality of the fuel should be determined by calorific tests and analysis of the coal sample above referred to.

12. ANALYSES OF FLUE GASES.

For approximate determinations of the composition of the flue gases, the Orsat apparatus, or some modification thereof, should be employed. If momentary samples are obtained the analyses should be made as frequently as possible, say every 15 to 30 minutes, depending on the skill of the operator, noting at the time the sample is drawn, the furnace and firing conditions. If the sample drawn is a continuous one, the intervals may be made longer.

13. SMOKE OBSERVATIONS.

In tests of bituminous coals requiring a determination of the amount of smoke produced, observations should be made regularly throughout the trial at intervals of five minutes (or if necessary every minute), noting at the same time the furnace and firing conditions.

14. CALCULATION OF RESULTS.

The methods to be followed in expressing and calculating those results which are not self-evident are explained as follows:

(a) *Efficiency.* The "efficiency of boiler, furnace, and grate" is the relation between the heat absorbed per pound of coal fired and the calorific value of one pound of coal.

The "efficiency of boiler and furnace" is the relation between the heat absorbed per pound of combustible burned, and the calorific value of one pound of combustible. This expression of efficiency furnishes a means for comparing one boiler and furnace with another, when the losses of unburned coal due to grates, cleanings, etc., are eliminated.

The "combustible burned" is determined by subtracting from the weight of coal supplied to the boiler, the moisture in the coal, the weight of ash and unburned coal withdrawn from the furnace and ashpit, and the weight of dust, soot, and refuse, if any, withdrawn from the tubes, flues, and combustion

chambers, including ash carried away in the gases, if any, determined from the analyses of coal and ash. The "combustible" used for determining the calorific value is the weight of coal less the moisture and ash found by analysis.

The "heat absorbed" per pound of coal, or combustible, is calculated by multiplying the equivalent evaporation from and at 212 degrees per pound of coal or combustible by 970.4.

(b) *Corrections for Moisture in Steam.* When the percentage is less than 2 per cent it is sufficient merely to deduct the percentage from the weight of water fed. If the percentage is greater than 2 per cent or if extreme accuracy is required, the factor of correction equals

$$Q + P\frac{(T - h)}{(H - h)},$$

in which Q is the quality of the steam (one minus the decimal representing the percentage of moisture), P the proportion of moisture, T the total heat of water at the temperature of the steam, h the total heat of the feed water, and H the total heat of saturated steam of the given temperature.

(c) *Correction for Live Steam, if Any, used for Aiding Combustion.* If live steam is admitted into the furnace or ashpit for producing blast, injecting fuel, or aiding combustion, it is to be deducted from the total evaporation, and the net evaporation used in the various calculations.

(d) *Equivalent Evaporation.* The equivalent evaporation from and at 212 degrees is obtained by multiplying the weight of water evaporated, corrected for moisture in steam, by the "factor of evaporation." The latter equals

$$\frac{H - h}{970.4},$$

in which H and h are respectively the total heat of saturated steam and of the feed water entering the boiler. When the steam is superheated, the total heat of the steam is that of saturated steam plus the product of the number of degrees of superheating by the specific heat of the steam.

Unless otherwise provided, a combined boiler and superheater should be treated as one unit, and the equivalent of the work done by the superheater should be included in the evaporative work of the boiler.

(e) *Heat Balance.* The "heat balance," or approximate distribution of the calorific value of the coal or combustible among the several items of heat utilized and heat lost, should be obtained in cases where the flue gases have been analyzed and a complete analysis made of the coal.

The loss due to moisture in the coal is found by multiplying the total heat of one pound of superheated steam at the temperature of the escaping gases, calculated from the temperature of the air in the boiler room, by the proportion of moisture.

The loss due to moisture formed by the burning of hydrogen is obtained by multiplying the total heat of one pound of superheated steam at the temperature of the escaping gases, calculated from the temperature of the air in the boiler room, by the proportion of the hydrogen, determined from the analysis of the coal, and multiplying the result by 9.

The loss due to heat carried away in the dry gases is found by multiplying the weight of gas per pound of coal or combustible by the elevation of temperature of the gases above the temperature of the boiler room, and by the

specific heat of the gases (0.24). The weight of gas referred to is obtained by finding the weight of dry gas per pound of carbon burned, using the formula

$$\frac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)},$$

in which CO_2, CO, O, and N are expressed in percentages by volume, and multiplying this result by the proportion borne by the carbon burned to the whole amount of coal or combustible as determined from the results of the analysis of the coal, ash, and refuse.

The loss due to incomplete combustion of carbon is found by first obtaining the proportion borne by the carbon monoxide in the gases to the sum of the carbon monoxide and carbon dioxide, and then multiplying this proportion by the proportion of carbon in the coal or combustible, and finally multiplying the product by 10,150, which is the number of heat units generated by burning to carbon dioxide one pound of carbon contained in carbon monoxide.

The loss due to combustible matter in the ash and refuse is found by multiplying the proportion that this combustible bears to the whole amount of coal or combustible, by its calorific value per pound. For most purposes it is sufficient to assume the latter to be 14,600 B.t.u., the same as that of carbon.

The loss due to moisture in the air is determined by multiplying the weight of such moisture per pound of coal or combustible by the elevation of temperature of the flue gases above the temperature of the boiler room and by 0.47. The weight of moisture is found by multiplying the weight of air per pound of coal or combustible by the moisture in one pound of air determined from readings of the wet and dry-bulb thermometer.

(*f*) *Total Heat of Combustion of Coal, by Analysis.* The total heat of combustion may be computed from the results of the ultimate analysis by using the formula

$$14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right) + 4000\,S,$$

in which C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur, respectively.

(*g*) *Air for Combustion.* The quantity of air used may be calculated by the formulæ:

$$\text{Lb. of air per lb. of carbon} = \frac{3.032\,N}{CO_2 + CO},$$

in which N, CO_2, and CO are the percentages of dry gas obtained by analysis, and

Lb. of air per lb. of coal = lb. air per lb. C × per cent C in the coal.

The ratio of the air supply to that theoretically required for complete combustion is $\frac{N}{N - 3.782\,O}$.

15. DATA AND RESULTS.

The data and results should be reported in accordance with either the Short Form or the Complete Form printed below, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

16. CHART.

In trials having for an object the determination and exposition of the complete boiler performance, the entire log of readings and data should be plotted on a chart and represented graphically. See Fig. 606.

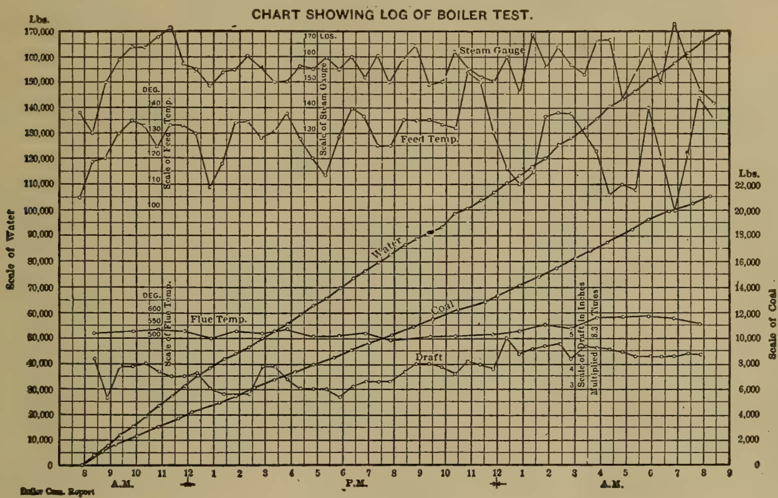


FIG. 606.

17. TESTS WITH OIL AND GAS FUELS.

Tests of boilers using oil or gas for fuel should accord with the rules here given, excepting as they are varied to conform to the particular characteristics of the fuel. The duration in such cases may be reduced, and the "flying" method of starting and stopping employed.

The table of data and results should contain items stating character of furnace and burner, quality and composition of oil or gas, temperature of oil, pressure of steam used for vaporizing, and quantity of steam used both for vaporizing and for heating.

TABLE 1. DATA AND RESULTS OF EVAPORATIVE TEST, SHORT FORM, CODE OF 1912.

(1) Test of.................... boiler located at..............................
to determine............... conducted by ....
(2) Kind of furnace..
(3) Grate surface..................... ....sq. ft.
(4) Water-heating surface..sq. ft.
(5) Superheating surface..sq. ft.
(6) Date...
(7) Duration.................................. ....hr.
(8) Kind and size of coal...

Average Pressures, Temperatures, etc.

(9) Steam pressure by gage .... lb.
(10) Temperature of feed water entering boiler .... deg.
(11) Temperature of escaping gases leaving boiler .... deg.
(12) Force of draft between damper and boiler .... in.
(13) Percentage of moisture in steam, or number of degrees of superheating .... per cent or deg.

Total Quantities.

(14) Weight of coal as fired* .... lb.
(15) Percentage of moisture in coal .... per cent.
(16) Total weight of dry coal consumed .... lb.
(17) Total ash and refuse .... lb.
(18) Percentage of ash and refuse in dry coal .... per cent.
(19) Total weight of water fed to the boiler † .... lb.
(20) Total water evaporated, corrected for moisture in steam .... lb.
(21) Total equivalent evaporation from and at 212 degrees .... lb.

Hourly Quantities and Rates.

(22) Dry coal consumed per hour .... lb.
(23) Dry coal per square foot of grate surface per hour .... lb.
(24) Water evaporated per hour corrected for quality of steam .... lb.
(25) Equivalent evaporation per hour from and at 212 degrees .... lb.
(26) Equivalent evaporation per hour from and at 212 degrees per square foot of water-heating surface .... lb.

Capacity.

(27) Evaporation per hour from and at 212 degrees (same as Line 25) .... lb.
(28) Boiler horse power developed (Item 27 $\div$ $34\frac{1}{2}$) .... bl.h.p.
(29) Rated capacity, in evaporation from and at 212 degrees per hour ...... lb.
(30) Rated boiler horse power .... bl.h.p.
(31) Percentage of rated capacity developed .... per cent.

Economy Results.

(32) Water fed per pound of coal fired (Item 19 $\div$ Item 14) .... lb.
(33) Water evaporated per pound of dry coal (Item 20 $\div$ Item 16) .... lb.
(34) Equivalent evaporation from and at 212 degrees per pound of dry coal (Item 21 $\div$ Item 16) .... lb.
(35) Equivalent evaporation from and at 212 degrees per pound of combustible [Item 21 $\div$ (Item 16 $-$ Item 17)] .... lb.

Efficiency.

(36) Calorific value of 1 pound of dry coal .... B.t.u.
(37) Calorific value of 1 pound of combustible .... B.t.u.
(38) Efficiency of boiler, furnace, and grate $\left[100 \times \frac{\text{Item 34} \times 970.4}{\text{Item 36}}\right]$ .... per cent.
(39) Efficiency of boiler and furnace $\left[100 \times \frac{\text{Item 35} \times 970.4}{\text{Item 37}}\right]$ .... per cent.

* The term "as fired" means actual conditions, including moisture, corrected for estimated difference in weight of coal on the grate at beginning and end.

† Corrected for inequality of water level and steam pressure at beginning and end.

Cost of Evaporation.

(40) Cost of coal per ton of .... pounds delivered in boiler room..........dollars
(41) Cost of coal required for evaporating 1000 pounds of water from and at 212 degrees..........dollars

TABLE 2. DATA AND RESULTS OF EVAPORATIVE TEST, COMPLETE FORM, CODE OF 1912.

(1) Test of....................boiler located at..........................
to determine..............., conducted by........................

Dimensions, Proportions, etc.

(2) Number and kind of boilers....................................
(3) Kind of furnace....................................
(4) Grate surface........width........length........area..............sq. ft.
(5) Approximate width of air spaces in grate............................in.
(6) Proportion of air space to whole grate surface....................per cent.
(7) Water-heating surface......................................sq. ft.
(8) Superheating surface......................................sq. ft.
(9) Ratio of water-heating surface to grate surface......................to 1
(10) Ratio of minimum draft area to grate surface....................1 to ....
(11) Date....................................
(12) Duration......................................hr.
(13) Kind of coal....................................
(14) Size of coal....................................

Average Pressures, Temperatures, etc.

(15) Steam pressure by gage......................................lb.
(16) Barometric pressure......................................lb.
(17) Force of draft at dampers of individual boilers........................in.
(18) Force of draft in main flue near boilers..............................in.
(19) Force of draft in main flue between economizer and chimney............in.
(20) Force of draft in furnaces......................................in.
(21) Force of blast in ashpits......................................in.
(22) State of weather....................................
(23) Temperature of external air......................................deg.
(24) Temperature of fireroom......................................deg.
(25) Temperature of steam......................................deg.
(26) Normal temperature of saturated steam..............................deg.
(27) Temperature of feed water entering flue heater or economizer ....deg.
(28) Temperature of feed water leaving heater or economizer and entering boilers......................................deg.
(29) Temperature of gases leaving boilers..............................deg.
(30) Temperature of gases leaving economizer..............................deg.

Quality of Steam.

(31) Percentage of moisture in steam ....per cent.
(32) Number of degrees of superheating..............................deg.
(33) Quality of steam (dry steam = unity) ....

Total Quantities.

(34) Weight of coal as fired*..lb.
(35) Percentage of moisture in coal..............................per cent.
(36) Total weight of dry coal consumed...............................lb.
(37) Total ash and refuse...lb.
(38) Total combustible consumed (Line 36 − Line 37).......................lb.
(39) Percentage of ash and refuse in dry coal..........................per cent.
(40) Total weight of water fed to boiler....................................lb.
(41) Total water evaporated corrected for moisture in steam..................lb.
(42) Factor of evaporation, based on temperature of water entering boilers........
(43) Total equivalent evaporation from and at 212 degrees..................lb.

Hourly Quantities and Rates.

(44) Dry coal consumed per hour...lb.
(45) Combustible consumed per hour..lb.
(46) Dry coal per square foot of grate surface per hour.....................lb.
(47) Water evaporated per hour, corrected for quality of steam...............lb.
(48) Equivalent evaporation per hour from and at 212 degrees†...............lb.
(49) Equivalent evaporation per hour from and at 212 degrees per square foot of water-heating surface‡..lb.

Proximate Analysis of Coal.

(50) Fixed carbon..per cent.
(51) Volatile matter...per cent.
(52) Moisture..per cent.
(53) Ash...per cent.

100 per cent.

(54) Sulphur, separately determined.....................................per cent.

Ultimate Analysis of Dry Coal.

(55) Carbon (C)...per cent.
(56) Hydrogen (H)...per cent.
(57) Oxygen (O)...per cent.
(58) Nitrogen (N)...per cent.
(59) Sulphur (S)..per cent.
(60) Ash..per cent.

100 per cent.

(61) Moisture in sample of coal as received.............................per cent.

Analysis of Ash and Refuse.

(62) Carbon...per cent.
(63) Earthy matter..per cent.
  (*a*) SiO_2
  (*b*) Al_2O_3 and Fe_2O_3
  (*c*) MgO
  (*d*) CaO
(64) Temperature of fusion of ash...deg.

* The term " as fired " means actual conditions, including moisture, corrected for estimated difference in weight of coal on the grate at beginning and end.

† Corrected for inequality of water level and steam pressure at beginning and end.

‡ The symbol U. E., meaning Units of Evaporation, may be substituted for the expression, Equivalent water evaporated into dry steam from and at 212 deg.

Calorific Value.

(65) Calorific value of 1 pound of dry coal by calorimeter .... B.t.u.
(66) Calorific value of 1 pound of combustible by calorimeter .... B.t.u.
(67) Calorific value of 1 pound of dry coal by analysis .... B.t.u.
(68) Calorific value of 1 pound of combustible by analysis .... B.t.u.

Capacity.

(69) Evaporation per hour from and at 212 degrees (same as Line 48) .... lb.
(70) Boiler horse power developed (Line 69 ÷ $34\frac{1}{2}$) .... bl.h.p.
(71) Rated capacity per hour, from and at 212 degrees .... lb.
(72) Rated boiler horse power .... bl.h.p.
(73) Percentage of rated capacity developed .... per cent.

Economy Results.

(74) Water fed per pound of coal (Item 40 ÷ Item 34) .... lb.
(75) Water evaporated per pound of dry coal (Item 41 ÷ Item 36) .... lb.
(76) Equivalent evaporation from and at 212 degrees per pound of coal fired (Item 43 ÷ Item 34) .... lb.
(77) Equivalent evaporation from and at 212 degrees per pound of dry coal (Item 43 ÷ Item 36) .... lb.
(78) Equivalent evaporation from and at 212 degrees per pound of combustible (Item 43 ÷ Item 38) .... lb.

Efficiency.

(79) Efficiency of boiler, furnace, and grate $\left[100 \times \frac{\text{Item 77} \times 970.4}{\text{Item 65}}\right]$ .... per cent.

(80) Efficiency of boiler and furnace $\left[100 \times \frac{\text{Item 78} \times 970.4}{\text{Item 66}}\right]$ .... per cent.

Cost of Evaporation.

(81) Cost of coal per ton of .... pounds delivered in boiler room .... dollars
(82) Cost of coal required for evaporating 1000 pounds of water under observed conditions .... dollars
(83) Cost of coal required for evaporating 1000 pounds of water from and at 212 degrees .... dollars

Smoke Data.

(84) Percentage of smoke as observed .... per cent.
(85) Weight of soot per hour obtained from smoke meter ....

Methods of Firing.

(86) Kind of firing, whether spreading, alternate, or cooking ....
(87) Average thickness of fire ....
(88) Average intervals between firings for each furnace during time when fires are in normal condition ....
(89) Average interval between times of leveling or breaking up ....

Analysis of Dry Gases by Volume.

(90) Carbon dioxide (CO_2) .... per cent.
(91) Oxygen (O) .... per cent.
(92) Carbon monoxide (CO) .... per cent.
(93) Hydrogen and hydrocarbons .... per cent.
(94) Nitrogen, by difference (N) .... per cent.

100 per cent.

Heat Balance, Based on Dry Coal and Combustible.

	Dry Coal as Fired		Combustible Burned	
	B.t.u.	Per Cent	B.t.u.	Per Cent
(95) Heat absorbed by the boiler (Line 76 or 77 × 970.4).				
(96) Loss due to evaporation of moisture in coal				
(97) Loss due to heat carried away by steam formed by the burning of hydrogen				
(98) Loss due to heat carried away in the dry flue gases ..				
(99) Loss due to carbon monoxide				
(100) Loss due to combustible in ash and refuse				
(101) Loss due to heating moisture in air				
(102) Loss due to unconsumed hydrogen and hydro-carbons, to radiation, and unaccounted for				
(103) Total calorific value of 1 lb. of dry coal or combustible (Lines 65 and 66)		100		100

APPENDIX C.

RULES FOR CONDUCTING TESTS OF RECIPROCATING ENGINES.*

A.S.M.E. Code of 1912.

1. OBJECT AND PREPARATIONS.

Determine the object, take the dimensions, note the physical conditions not only of the engine but of all parts of the plant that are concerned in the determinations, examine for leakages, install the testing appliances, etc., as pointed out in Appendix A and prepare for the test accordingly.

2. APPARATUS AND INSTRUMENTS.

The apparatus and instruments required for a simple performance test of a steam engine, in which the steam consumption is determined by feed-water measurement, are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place).
(*b*) Graduated scales attached to the water glasses of the boilers
(*c*) Pressure gauges, vacuum gauges, and thermometers.
(*d*) A steam calorimeter.
(*e*) A barometer.
(*f*) Steam-engine indicators.
(*g*) A planimeter.
(*h*) A tachometer or other speed-measuring apparatus.
(*i*) A friction brake or dynamometer.

The determination of the heat and steam consumption of an engine by feed-water test requires the measurement of the various supplies of water fed to the boiler; that of the water discharged by separators and drips not returned to the boiler, and that of water and steam which escapes by leakage of the boiler and piping, all of these last being deducted from the total feed water measured.

To ascertain the consumption of heat, the various feed temperatures are taken and heat calculations made accordingly. If the conditions imposed by the particular method adopted for carrying on the test depart from the usual practice, as for example where a colder supply of feed water is used than the ordinary supply, a preliminary or subsequent

* Preliminary report of the Committee on Power Tests. (Jour. A.S.M.E., Nov., 1912.) Greatly abridged.

run should be made to ascertain the temperatures which obtain under the usual working conditions, and the heat measurements obtained under the test conditions appropriately corrected for such departures.

The steam consumed by steam-driven auxiliaries which are required for the operation of the engine should be included in the total steam from which the heat consumption is calculated and the quantity of steam thus used should be determined and reported.

3. OPERATING CONDITIONS.

Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial.

4. DURATION.

A test for heat or steam consumption, with substantially constant load, should be continued for such time as may be necessary to obtain a number of successive hourly records, during which the results are reasonably uniform. For a test involving the measurement of feed water for this purpose, five hours is sufficient duration. Where a surface condenser is used, and the measurement is that of the water discharged by the air pump, the duration may be somewhat shorter. In this case, successive half-hourly records may be compared and the time correspondingly reduced.

When the load varies widely at different times of the day, the duration should be such as to cover the entire period of variation.

The preliminary or subsequent trial for determining the working temperatures on a heat test, where the temperatures obtained under the test conditions depart from the usual temperatures, should be of such duration as may be required to secure working results.

5. STARTING AND STOPPING.

The engine and appurtenances having been set to work and thoroughly heated under the prescribed conditions of test, except in cases where the object is to obtain the performance under working conditions, note the water levels in the boilers and feed reservoir, take the time and consider this the starting time. Then begin the measurements and observations and carry them forward until the end of the period determined on. When this time arrives, the water levels and steam pressure should be brought as near as practicable to the same points as at the start. This being done, again note the time and consider it the stopping time of the test. If there are differences in the water levels, proper corrections are to be applied.

Where a surface condenser is used, the collection of water discharged by the air pump begins at the starting time, and the water is thereafter measured or weighed until the end of the test; no observations of the boilers being required.

6. RECORDS.

The general data should be recorded as pointed out in Appendix A, under the heading Records. Half-hourly readings of the instruments are sufficient, excepting where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes it necessary. Mark on each card the cylinder and the end on which it was taken, also the time of day. Record on one card of each set the readings of the pressure gauges concerned, taken at the same time. These records should subsequently be entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are worked up.

7. CALCULATION OF RESULTS.

(*a*) *Dry Steam.* The quantity of dry steam consumed when there is no superheating is determined by deducting the moisture found by calorimeter test from the total amount of feed water (the latter being corrected for leakages) or from the amount of air-pump discharge, as the case may be.

When there is superheating the dry steam is found by multiplying the weight of superheated steam by the factor

$$1 + \frac{C\,(T - t)}{H - h},$$

in which

C = specific heat of superheated steam at observed pressure and temperature.
T = temperature of superheated steam.
t = temperature of saturated steam.
H = total heat of saturated steam of observed pressure.
h = total heat of feed water.

(*b*) *Heat Consumption.* The number of heat units consumed by the engine is found by multiplying the weight of feed water consumed, corrected for leakages, by the total heat of the steam above the working feed temperature, and multiplying the product by a factor of correction expressing the quality of the steam.

If the steam contains moisture, this factor equals

$$Q + P\,\frac{T - h}{H - h},$$

in which Q is the quality of the steam (one minus the decimal representing the percentage of moisture), P the proportion of moisture, T the total heat

of water at the temperature of the steam, h the total heat of the feed water, and H the total heat of saturated steam.

If the steam is superheated, the factor is that given above under (*a*) Dry Steam.

If there are a number of sources of feed-water supply, the corresponding heat units should be determined for each supply and the various quantities added together.

The British standard of heat consumption is based on a feed-water temperature assumed to be that of the temperature of saturated steam corresponding to the observed back pressure (whether this is above or below the atmosphere), plus the temperature due to heat derived from jacket or reheater drips. It does not include the heat consumed by any auxiliaries, except jackets and reheaters.

(*c*) *Indicated Horse Power.* In a single double-acting cylinder the indicated horse power is found by using the formula

$$\frac{PLAN}{33{,}000},$$

in which P represents the average mean effective pressure in pounds per square inch measured from the indicator diagrams, L the length of stroke in feet, A the area of the piston less one-half the area of the piston rod, or the mean area of the rod if it passes through both cylinder heads, in square inches, and N the number of single strokes per minute.

Where extreme accuracy is required, the power developed by each side of the piston may be determined and the results added together.

(*d*) *Brake Horse Power.* The brake horse power is found by multiplying the net weight on the brake arm (the gross weight minus the weight when the brake is entirely free) in pounds, the circumference of the circle passing through the bearing point at the end of the brake arm, in feet, and the number of revolutions of the brake shaft per minute, and dividing the product by 33,000.

(*e*) *Electrical Horse Power.* The electrical horse power for a direct-connected generator is found by dividing the output at the bus-bar, expressed in kilowatts, by the decimal 0.746. For alternating-current systems the net output is to be used, being the total output less that consumed for excitation.

(*f*) *Efficiency.* The efficiency is expressed by the thermal efficiency ratio, which is found by dividing the quantity 2545 by the number of heat units consumed per h.p. hr., either indicated or brake.

(*g*) *Steam accounted for by Indicator Diagrams.* The steam accounted for, expressed in pounds per i.h.p. per hour, may readily be found by using the formula

$$\frac{13{,}750}{\text{m.e.p.}}\left[(C+E)\,Wc-(H+E)\,Wh\right],$$

in which

m.e.p. = mean effective pressure.
C = proportion of stroke completed at cut-off or release.
E = proportion of clearance.
H = proportion of stroke uncompleted at compression.
Wc = weight of 1 cubic foot steam at cut-off or release pressure.
Wh = weight of 1 cubic foot steam at compression pressure.

The points of cut-off release and compression, referred to, are indicated in Fig. 607.*

In multiple-expansion engines the mean effective pressure to be used in the above formula is the combined m.e.p. referred to the cylinder under consideration. In a compound engine the combined m.e.p. for the h.p. cylinder is the sum of the actual m.e.p. of the h.p. cylinder and that of the l.p. cylinder multiplied by the cylinder ratio. Likewise the combined m.e.p. for the l.p. cylinder is the sum of the actual m.e.p. of the l.p. cylinder and the m.e.p. of the h.p. cylinder divided by the cylinder ratio.

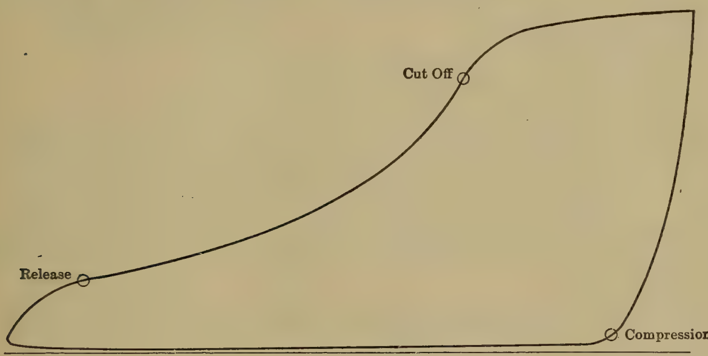


FIG. 607. Points where "Steam accounted for by Indicator" is Computed.

(*h*) *Cut-Off and Ratio of Expansion.* To find the percentage of cut-off, or what may best be termed the "commercial cut-off," the following rule should be observed:

> Through the point of maximum pressure during admission draw a line parallel to the atmospheric line. Through a point on the expansion line where the cut-off is complete, draw a hyperbolic curve. The intersection of these two lines is the point of commercial cut-off, and the proportion of cut-off is found by dividing the length measured on the diagram up to this point by the total length.

To find the ratio of expansion divide the volume corresponding to the piston displacement, including clearance, by the volume of the steam at the commercial cut-off, including clearance.

In a multiple-expansion engine the ratio of expansion is found by dividing the volume of the l.p. cylinder, including clearance, by the volume of the h.p. cylinder at the commercial cut-off, including clearance.

8. DATA AND RESULTS.

The data and results should be reported in accordance with either the Short Form or Complete Form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

* Reproduced from Report of Committee on Standardizing Engine Tests, Fig. 122, Trans. Am. Soc. M. E., vol. 24, p. 744.

TABLE 1. DATA AND RESULTS OF HEAT AND FEED-WATER TESTS OF STEAM ENGINE.

SHORT FORM, CODE OF 1912.

(1) Test of....................engine located at.............................
to determine................conducted by........................
(2) Type and class of engine and auxiliaries...................................

(3) Dimensions of main engine:	1st Cyl.	2d Cyl.	3d Cyl.
(a) Diameter of cylinder...................in.	......	......	......
(b) Stroke of piston.......................ft.	......	......	......
(c) Diameter of piston rod each end.........in.	......	......	......
(d) Average clearance................per cent	......	......	......
(e) Cylinder ratio..........................	......	......	......
(f) Horse-power constant for 1 lb. m.e.p. and 1 r.p.m..........................	......	......	......

(4) Dimensions and type of auxiliaries.......................................
(5) Date...
(6) Duration...hr.

Average Pressures and Temperatures.

(7) Pressure in steam pipe near throttle by gauge.........................lb.
(8) Barometric pressure of atmosphere in in. of mercury...................in.
(9) Pressure in receivers by gauge..lb.
(10) Vacuum in condenser in inches of mercury.............................in.
(11) Pressure in jackets and reheaters by gauge...........................lb.
(12) Temperature of main supply of feed water............................deg.
(13) Temperature of additional supplies of feed water....................deg.

Total Quantities.

(14) Total water fed to boilers from main source of supply................lb.
(15) Total water fed from additional supplies.............................lb.
(16) Total water fed to boilers from all sources..........................lb.
(17) Moisture in steam or superheating near throttle.............per cent or deg.
(18) Factor of correction for quality of steam................................
(19) Total dry steam consumed for all purposes............................lb.

Hourly Quantities.

(20) Water fed from main source of supply.................................lb.
(21) Water fed from additional supplies...................................lb.
(22) Total water fed to boilers per hour..................................lb.
(23) Total dry steam consumed per hour....................................lb.
(24) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant...lb.
(25) Net dry steam consumed per hour by engine and auxiliaries.............lb.
(26) Net dry steam consumed per hour:
(a) By engine alone.......................................lb.
(b) By auxiliaries..lb.

HEAT DATA.

(27) Heat units per pound of dry steam, based on temperature of Line 12 . . . . . . B.t.u.
(28) Heat units per pound of dry steam, based on temperature of Line 13 . . . . . . B.t.u.
(29) Heat units consumed per hour, main supply of feed . . . . B.t.u.
(30) Heat units consumed per hour, additional supplies of feed . . . . B.t.u.
(31) Total heat units consumed per hour for all purposes . . . . B.t.u.
(32) Loss of heat per hour due to leakage of plant, drips, etc. . . . . B.t.u.
(33) Net heat units consumed per hour:
(*a*) By engine and auxiliaries . . . . B.t.u.
(*b*) By engine alone . . . . B.t.u.
(*c*) By auxiliaries . . . . B.t.u.

INDICATOR DIAGRAMS.

	1st Cyl.	2d Cyl.	3d Cyl.
(34) Commercial cut-off in per cent of stroke	. . . . . .	. . . . . .	. . . . . .
(35) Initial pressure in lb. per sq. in. above atmosphere	. . . . . .	. . . . . .	. . . . . .
(36) Back pressure at lowest point above or below atmosphere in lb. per sq. in.	. . . . . .	. . . . . .	. . . . . .
(37) Mean effective pressure in lb. per sq. in.	. . . . . .	. . . . . .	. . . . . .
(38) Steam accounted for by indicator in lb. per i.h.p. per hour:			
(*a*) Near cut-off	. . . . . .	. . . . . .	. . . . . .
(*b*) Near release	. . . . . .	. . . . . .	. . . . . .

SPEED.

(39) Revolutions per minute . . . . rev.
(40) Piston speed in feet per minute . . . . ft.

POWER.

(41) Indicated horse power developed by main-engine cylinders:
1st cylinder . . . . i.h.p.
2d cylinder . . . . i.h.p.
3d cylinder whole engine . . . . i.h.p.
Whole engine . . . . i.h.p.
(42) Brake horse power . . . . br.h.p.

ECONOMY RESULTS.

(43) Heat units consumed by engine and auxiliaries per hour:
(*a*) Per indicated horse power . . . . B.t.u.
(*b*) Per brake horse power . . . . B.t.u.
(44) Dry steam consumed per indicated horse power per hour:
(*a*) By engine and auxiliaries . . . . lb.
(*b*) By main engine alone . . . . lb.
(*c*) By auxiliaries . . . . lb.
(45) Dry steam consumed per brake horse power per hour:
(*a*) By engine and auxiliaries . . . . lb.
(*b*) By main engine alone . . . . lb.
(*c*) By auxiliaries . . . . lb.
(46) Percentage of steam used by main-engine cylinders accounted for by indicator diagrams:
(*a*) Near cut-off . . . . per cent.
(*b*) Near release . . . . per cent.

SAMPLE DIAGRAMS.

TABLE 2. DATA AND RESULTS OF STEAM-ENGINE TEST, COMPLETE FORM, CODE OF 1912.

(1) Test of.....................engine located at..........................
to determine.................conducted by.........................

(2) Type of engine (simple, compound, or other multiple expansion; condensing or non-condensing)..

(3) Class of engine (mill, marine, electric, etc.)..............................

(4) Rated power of engine..

(5) Name of builders..

(6) Number and arrangement of cylinders of engine; how lagged; type of condenser..

(7) Type of valves..

(8) Type of boiler...

(9) Kind and type of auxiliaries (air pump, circulating pump, feed pump; jackets, heaters, etc.)...

(10) Dimensions of engine:	1st Cyl.	2d Cyl.	3d Cyl.
(*a*) Single or double acting....................	......	......	......
(*b*) Cylinder dimensions:			
Bore, in..............................	......	......	......
Stroke, ft..............................	......	......	......
Diameter of piston rod, in...............	......	......	......
Diameter of tail rod, in.................	......	......	......
(*c*) Clearance in per cent of volume displaced by piston per stroke:			
Head end, per cent.....................	......	......	......
Crank end, per cent.....................	......	......	......
Average, per cent......................	......	......	......
(*d*) Surface in square feet (average):			
Barrel of cylinder, square feet...........	......	......	......
Cylinder heads, square feet.............	......	......	......
Clearance and ports, square feet.........	......	......	......
Ends of piston, square feet..............	......	......	......
(*c*) Jacket surfaces or internal surfaces of cylinder heated by jackets, in square feet:			
Barrel of cylinder, square feet...........	......	......	......
Cylinder heads, square feet.............	......	......	......
Clearance and ports, square feet.........	......	......	......
Receiver jackets, square feet............	......	......	......
(*g*) Horse power constant for 1 lb. m.e.p. and 1 r.p.m..........................	......	......	......

(11) Dimensions of boilers:

(*a*) Number...

(*b*) Total grate surface.....................................sq. ft.

(*c*) Total water-heating surface.............................sq. ft.

(*d*) Total steam-heating surface.............................sq. ft.

(12) Dimensions of auxiliaries:
 (*a*) Air pump . . . .
 (*b*) Circulating pump . . . .
 (*c*) Feed pumps . . . .
 (*d*) Heaters . . . .
(13) Dimensions of condenser . . . .
(14) Dimensions of electric or other machinery driven by engine . . . .
(15) Date . . . .
(16) Duration . . . . hr.

Average Pressures and Temperatures.

(17) Steam pressure at boiler by gauge . . . . lb.
(18) Steam-pipe pressure near throttle, by gauge . . . . lb.
(19) Barometric pressure of atmosphere in lb. per sq. in. . . . . lb.
(20) Pressure in first receiver by gauge . . . . lb.
(21) Pressure in second receiver by gauge . . . . lb.
(22) Vacuum in condenser:
 (*a*) In in. of mercury . . . . in.
 (*b*) Corresponding total pressure . . . . lb.
(23) Pressure in steam jacket by gauge . . . . lb.
(24) Pressure in reheater by gauge . . . . lb.
(25) Superheat in steam leaving first receiver . . . . deg.
(26) Superheat in steam leaving second receiver . . . . deg.
(27) Temperature of main supply of feed water to boilers . . . . deg.
(28) Temperature of additional supplies of feed water . . . . deg.
(29) Ideal feed-water temperature corresponding to the pressure of the steam in the exhaust pipe, allowance being made for heat derived from jacket or reheater drips (British Standard) . . . . deg.
(30) Temperature of injection or circulating water entering condenser . . . . deg.
(31) Temperature of injection or circulating water leaving condenser . . . . deg.
(32) Temperature of air in engine room . . . . deg.

Total Quantities.

(33) Water fed to boilers from main source of supply . . . . lb.
(34) Water fed from additional supplies . . . . lb.
(35) Total water fed to boilers from all sources . . . . lb.
(36) Moisture in steam or superheating near throttle . . . . per cent or deg.
(37) Factor of correction for quality of steam, dry steam being unity . . . . lb.
(38) Total dry steam consumed for all purposes . . . .

Hourly Quantities.

(39) Water fed from main source of supply . . . . lb.
(40) Water fed from additional supplies . . . . lb.
(41) Total water fed to boilers per hour . . . . lb.
(42) Total dry steam consumed per hour . . . . lb.
(43) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant . . . . lb.
(44) Net dry steam consumed per hour by engine and auxiliaries . . . . lb.

(45) Dry steam consumed per hour:
- (*a*) Main engine alone .... lb.
- (*b*) Jackets and reheaters .... lb.
- (*c*) Air pump .... lb.
- (*d*) Circulating pump .... lb.
- (*e*) Feed-water pump .... lb.
- (*f*) Other auxiliaries .... lb.

(46) Injection or circulating water supplied condenser per hour .... cu. ft.

Heat Data.

(47) Heat units per pound of dry steam, based on temperature of Line 27 ...... B.t.u.

(48) Heat units per pound of dry steam, based on temperature of Line 28 ...... B.t.u.

(49) Heat units consumed per hour, main supply of feed .... B.t.u.

(50) Heat units consumed per hour, additional supplies of feed .... B.t.u.

(51) Total heat units consumed per hour for all purposes .... B.t.u.

(52) Loss of heat per hour due to leakage of plant, drips, etc. .... B.t.u.

(53) Net heat units consumed per hour:
- (*a*) By engine and auxiliaries .... B.t.u.
- (*b*) By engine alone .... B.t.u.
- (*c*) By auxiliaries .... B.t.u.

(54) Heat units consumed per hour by the engine alone, reckoned from temperature given in Line 29 (British Standard) .... B.t.u.

Indicator Diagrams.

	1st Cyl.	2d Cyl.	3d Cyl.
(55) Commercial cut-off in per cent of stroke	......	......	......
(56) Initial pressure in lb. per sq. in. above atmosphere.	......	......	......
(57) Back-pressure at mid-stroke above or below atmosphere in lb. per sq. in.	......	......	......
(58) Mean effective pressure in lb. per sq. in.	......	......	......
(59) Equivalent mean effective pressure in lb. per sq. in.:			
(*a*) Referred to first cylinder	......	......	......
(*b*) Referred to second cylinder	......	......	......
(*c*) Referred to third cylinder	......	......	......
(60) Pressures and percentages used in computing the steam accounted for by the indicator diagrams, measured to points on the expansion and compression curves:			
Pressure above zero in lb. per sq. in.:			
(*a*) Near cut-off	......	......	......
(*b*) Near release	......	......	......
(*c*) Near beginning of compression	......	......	......
Percentage of stroke at points where pressures are measured:			
(*d*) Near cut-off	......	......	......
(*e*) Near release	......	......	......
(*f*) Near beginning of compression	......	......	......
(61) Aggregate m.e.p. in lb. per sq. in. referred to each cylinder given in heading	......	......	......
(62) Mean back pressure above zero, lb. per sq. in.	......	......	......

(63) Steam accounted for in lb. per indicated horse power per hour: 1st Cyl. 2d Cyl. 3d Cyl.

(a) Near cut-off........................ ....

(b) Near release........................ ....

(64) Ratio of expansion........................ ....

(65) Mean effective pressure of ideal diagram.............................lb.

(66) Diagram factor..

SPEED.

(67) Revolutions per minute...rev.

(68) Piston speed per minute..ft.

(69) Variation of speed between no load and full load.......................rev.

(70) Fluctuation of speed on suddenly changing from full load to no load, measured by the increase in the revolutions due to the change ....rev.

POWER.

(71) Indicated horse power developed by main engine:

1st cylinder..i.h.p.

2d cylinder...i.h.p.

3d cylinder...i.h.p.

Whole engine..i.h.p.

(72) Brake horse power..br.h.p.

(73) Friction i.h.p. by diagrams, no load on engine, computed for average speed . i.h.p.

(74) Difference between Lines 71 and 72....................................h.p.

(75) Percentage of i.h.p. of main engine lost in friction.................per cent.

(76) Power developed by auxiliaries*.......................................i.h.p.

ECONOMY RESULTS.

(77) Heat units consumed per indicated horse power per hour:†

(a) By engine and auxiliaries.....................................B.t.u.

(b) By engine alone...B.t.u.

(78) Heat units consumed per brake horse power per hour:

(a) By engine and auxiliaries.....................................B.t.u.

(b) By engine alone...B.t.u.

(79) Heat units consumed by engine per hour, corresponding to ideal temperature of feed water given in Line 29, per indicated horse power (British Standard)...B.t.u.

(80) Dry steam consumed per i.h.p. per hour:

(a) By engine and auxiliaries.......................................lb.

(b) By main engine alone..lb.

(c) By auxiliaries..lb.

* These are not included in the power developed by the main engine.

† The h.p. on which the economy and efficiency results are based are those of the main engine given in Line 71.

NOTE: Both the Short Form and Complete Form here given refer to a steam engine used for general service.

For an engine driving an electric generator the form should be enlarged to include the electrical data, embracing the average voltage, number of amperes each phase, number of watts, number of watt-hours, average power factor, etc.; and the economy results based on the electrical output embracing the heat units and steam consumed per electric h.p. per hour and per kw.-hr., together with the efficiency of the generator. See table for Steam Turbine Code, Part 6.

Likewise, in a marine engine having a shaft dynamometer, the form should include the data obtained from this instrument, in which case the brake h.p. becomes the shaft h.p.

(81) Dry steam consumed per brake h.p. per hour:
(*a*) By engine and auxiliaries .... lb.
(*b*) By main engine alone .... lb.
(*c*) By auxiliaries .... lb.

(82) Percentage of steam used by main engine cylinders accounted for by indicator diagrams.

	1st Cyl.	2d Cyl.	3d Cyl.
(*a*) Near cut-off	......	......	......
(*b*) Near release	......	......	......

Efficiency Results.

(83) Thermal efficiency ratio for engine and auxiliaries:
(*a*) Per indicated horse power .... per cent.
(*b*) Per brake horse power .... per cent.

(84) Thermal efficiency ratio for engine alone:
(*a*) Per indicated horse power .... per cent.
(*b*) Per brake horse power .... per cent.

(85) Ratio of economy of engine to that of an ideal engine working with the Rankine cycle .... per cent.

Work Done Per Heat Unit.

(86) Ft.-lb. of net work per B.t.u. consumed by engine and auxiliaries (1,980,000 ÷ Line 78*a*) .... ft.-lb.

APPENDIX D.

RULES FOR CONDUCTING TESTS OF STEAM TURBINES AND TURBO-GENERATORS.*

A.S.M.E. Code of 1912.

1. OBJECT AND PREPARATIONS.

Determine the object, take the dimensions, note the physical conditions not only of the turbine but of the entire plant concerned, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions in Appendix A, and prepare for the test accordingly.

2. APPARATUS AND INSTRUMENTS.

The apparatus and instruments required for a simple performance test of a steam turbine or turbo-generator, in which the steam consumption is determined by feed-water measurement, are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place).
(*b*) Graduated scales attached to the water glasses of the boilers.
(*c*) Pressure gauges, vacuum gauges, and thermometers.
(*d*) A steam calorimeter.
(*e*) A barometer.
(*f*) A tachometer or other speed-measuring apparatus.
(*g*) A friction brake or dynamometer.
(*h*) Voltmeters, ammeters, wattmeters, and watt-hour meters for the electrical measurements in the case of a turbo-generator.

The determination of the heat and steam consumption of a turbine or turbo-generator should conform to the same methods as those described in the Steam Engine Code.

The steam consumed by steam-driven auxiliaries required for the operation of a turbine should be included in the total steam from which the heat consumption is calculated the same as in the case of the steam engine.

3. OPERATING CONDITIONS.

Determine what the operating conditions should be to conform to the object in view and see that they prevail throughout the trial.

* Preliminary Report of Committee on Power Tests. (Jour. A.S.M.E., Nov., 1912.) Somewhat abridged.

4. DURATION.

5. STARTING AND STOPPING.

6. RECORDS.

7. CALCULATION OF RESULTS.

The rules pertaining to the subjects Duration, Starting and Stopping, Records, and Calculation of Results are identically the same as those given under the respective headings in the Steam Engine Code, with the single exception of the matter relating to indicator diagrams and results computed therefrom; and reference may be made to that code for the directions required in these particulars.

8. DATA AND RESULTS.

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view:

DATA AND RESULTS OF STEAM TURBINE OR TURBO-GENERATOR TESTS.

CODE OF 1912.

(1) Test of .... turbine located at ....
to determine ...., conducted by ....
(2) Type of turbine and class of service ....
(3) Type of generator, kind of current, etc. ....
(4) Rated power of turbine ....
(5) Type of boiler ....
(6) Kind and type of auxiliaries (air pumps, circulating pumps, feed pumps, etc.)
(7) Dimensions of turbine or turbo-generator ....
(8) Dimensions of boilers ....
(9) Dimensions of auxiliaries ....
(10) Dimensions of condenser ....
(11) Date ....
(12) Duration ....hr.

Average Pressures and Temperatures.

(13) Steam-pipe pressure near throttle, by gauge ....lb.
(14) Steam-chest pressure by gauge ....lb.
(15) Barometric pressure of atmosphere in lb. per sq. in. ....lb.
(16) Vacuum in condenser:
(*a*) In inches of mercury ....in.
(*b*) Corresponding absolute pressure ....lb.
(17) Exhaust-chamber pressure (absolute) ....lb.
(18) Temperature of main supply of feed water to boilers ....deg.
(19) Temperature of additional supplies of feed water ....deg.
(20) Temperature of injection or circulating water entering condenser ....deg.
(21) Temperature of injection or circulating water leaving condenser ....deg.

Total Quantities.

(22) Water fed to boilers from main source of supply . . . . lb.
(23) Water fed from additional supplies . . . . lb.
(24) Total water fed to boilers from all sources . . . . lb.
(25) Moisture in steam or superheating near throttle . . . . per cent or deg.
(26) Factor of correction for quality of steam, dry steam being unity . . . .
(27) Total dry steam consumed for all purposes . . . . lb.

Hourly Quantities.

(28) Water fed from main source of supply . . . . lb.
(29) Water fed from additional supplies . . . . lb.
(30) Total water fed to boilers per hour . . . . lb.
(31) Total dry steam consumed per hour . . . . lb.
(32) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant . . . . lb.
(33) Net dry steam consumed per hour . . . . lb.
(34) Dry steam consumed per hour:
(*a*) By turbine . . . . lb.
(*b*) By auxiliaries . . . . lb.
(35) Injection or circulating water supplied condensers per hour . . . . cu. ft.

Heat Data.

(36) Heat units per pound of dry steam, based on temperature of Line 18 . . . . . . B.t.u.
(37) Heat units per pound of dry steam, based on temperature of Line 19 . . . . . . B.t.u.
(38) Heat units consumed per hour, main supply of feed . . . . B.t.u.
(39) Heat units consumed per hour, additional supplies of feed . . . . B.t.u.
(40) Total heat units consumed per hour for all purposes . . . . B.t.u.
(41) Loss of heat per hour due to leakage of plant, drips, etc. . . . . B.t.u.
(42) Heat units consumed per hour:
(*a*) By turbine and auxiliaries . . . . B.t.u.
(*b*) By turbine alone . . . . B.t.u.
(*c*) By auxiliaries . . . . B.t.u.

Electrical Data.

(43) Average volts, each phase . . . . volts.
(44) Average amperes, each phase . . . . amperes.
(45) Average kilowatts, first meter . . . . kw.
(46) Average kilowatts, second meter . . . . kw.
(47) Total kilowatt output . . . . kw.
(48) Power factor . . . .
(49) Output consumed by exciter . . . . kw.
(50) Net kilowatt output . . . . kw.

Speed.

(51) Revolutions per minute . . . . rev.
(52) Variation of speed between no load and full load . . . . rev.
(53) Fluctuation of speed on suddenly changing from full load to no load, measured by the increase in the revolutions due to the change . . . . rev.

Power.

(54) Brake horse power . . . . br.h.p.
(55) Electrical horse power . . . . h.p.

Economy Results.

(56) Heat units consumed by turbine and auxiliaries per brake h.-p. hr...... B.t.u.

(57) Dry steam consumed per brake h.-p. hr.:

(a) By turbine and auxiliaries.. lb.

(b) By turbine alone.. lb.

(c) By auxiliaries.. lb.

(58) Dry steam consumed per kw.-hr.:

(a) By turbine and auxiliaries.. lb.

(b) By turbine alone.. lb.

(c) By auxiliaries.. lb.

Efficiency Results.

(59) Thermal efficiency ratio per brake horse power..................... per cent.

(60) Ratio of economy of turbine to that of an ideal turbine working with the Rankine cycle..

Work Done per Heat Unit.

(61) Ft.-lb. of net work per B.t.u. consumed by turbine and auxiliaries (1,980,000 ÷ Line 56).. ft.-lb.

APPENDIX E.

RULES FOR CONDUCTING TESTS OF COMPLETE STEAM POWER PLANTS.*

A.S.M.E. Code of 1912.

1. OBJECT AND PREPARATIONS.

These rules are intended to apply to commercial tests of a complete plant to determine the number of pounds of fuel consumed per unit of work done in a unit of time. For tests of the component parts of a complete plant, such as boilers, engines, turbines, etc., rules may be found in there spective Codes applying to such cases.

Read the general instructions given in Appendix A. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

2. FUEL.

Determine the character of the fuel to be used according to the object in view. For further particulars reference may be made to the Boiler Code.

3. APPARATUS AND INSTRUMENTS.

The apparatus and instruments required for a simple performance test of a steam plant are:

(*a*) Platform scales for weighing coal and ashes.
(*b*) Coal calorimeter.
(*c*) Steam-engine indicators.
(*d*) A tachometer or other speed-measuring apparatus.
(*e*) Electrical instruments for determining the output of an electric plant.

If the test involves the determination of boiler performance, and engine or turbine performances, additional instruments should be used as pointed out in the respective Codes referring to such tests.

4. OPERATING CONDITIONS.

Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial.

* Preliminary Report of the Committee on Power Tests. (Jour. A.S.M.E., Nov., 1912.)

5. DURATION.

The duration of a plant test should be not less than one day of 24 hours, and preferably a full week of seven days, including Sunday.

In cases where the engine or turbine is in operation only a part of the day, the duration on which the results are computed, should be considered the length of time that the engine or turbine is in operation at its working speed.

6. STARTING AND STOPPING.

In a plant operating continuously, day and night, the times fixed for starting and stopping should follow the regular periods of cleaning the fires. The fires should be quickly cleaned and then burned low, say to a thickness of 4 inches. When this condition is reached the time should be noted as the starting time, and the thickness of each coal bed observed, as also the water levels and the steam pressure. Fresh coal should then be fired from that weighed for the test, the ashpits thoroughly cleaned, and the regular work of the test proceeded with. At the close of the test, following a regular cleaning, the fires should again be burned low, and when their condition has become the same as that observed at the beginning, the water levels and steam pressure also being the same, the time is observed and this time taken as the stopping time. If the water levels and steam pressure are not the same as at the beginning a suitable correction should be made by computation. The ashes and refuse are then hauled from the ashpits.

In a plant running only a part of the day, and during the balance of the day the fires are banked, the time selected for the beginning and end of the test should be that following the close of the day's run, when the fires have been burned low preparatory to cleaning and banking. The amount of live coal left on the grates under these circumstances is estimated at the beginning of the test, and the fires brought to the same condition, as near as may be, at the close of the test the next day. If the two quantities differ, a suitable correction is made in the weight of coal fired, as found by calculation.

7. RECORDS.

The general data should be recorded as pointed out in Appendix A, under the head of Records. Half-hourly readings of the various instruments concerned are usually sufficient, excepting where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes it necessary. Mark on each card the cylinder and the end on which it was taken, also the time of the day. Record on

one card of each set the readings of the pressure gauges concerned, taken at the same time. These records should subsequently be entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are the worked up.

8. SAMPLING AND DRYING COAL.

During the progress of the test the coal should be regularly sampled for the purpose of analysis and determination of moisture.

9. ASHES AND REFUSE.

The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed in a dry state, and, if desired, a representative sample should be obtained for proximate analysis and the determination of the amount of unburned carbon which it contains.

10. CALORIFIC TESTS AND ANALYSES OF COAL.

The quality of the fuel should be determined by calorific tests and analysis of the representative sample above referred to.

11. CALCULATION OF RESULTS.

The methods of calculating the indicated and electrical horse powers are the same as explained in the Steam Engine Code.

12. DATA AND RESULTS.

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

DATA AND RESULTS OF COMPLETE STEAM POWER PLANT TEST.

(1) Test of....................plant located at..............................
to determine................., conducted by.......................
(2) Type of engine or turbine and class of service............................
(3) Rated power of engine or turbine......................................
(4) Type of boilers..
(5) Kind and type of auxiliaries (air pump, circulating pump, and feed pump; jackets, heaters, etc.)..
(6) Dimensions of engine or turbine......................................
(7) Dimensions of boilers..
(8) Dimensions of auxiliaries...
(9) Dimensions of condenser..
(10) Date..
(11) Duration..hr.
(12) Length of time engine or turbine was in motion with throttle open..........hr.
(13) Length of time engine or turbine was running at normal speed............hr.
(14) Kind of coal...
(15) Size of coal...

Average Pressures and Temperatures.

(16) Steam pressure at boiler by gauge.......................................lb.
(17) Steam-pipe pressure near throttle, by gauge..............................lb.
(18) Barometric pressure of atmosphere in in. of mercury.....................in.
(19) Pressure in receiver by gauge...lb.
(20) Vacuum in condenser...in.
(21) Number of degrees of superheating, if any, near throttle...............deg.
(22) Temperature of feed water entering boilers..............................deg.

Total Quantities, Time, Etc.

(23) Total coal as fired *..lb.
(24) Moisture in coal..per cent.
(25) Total dry coal consumed...lb.
(26) Ash and refuse..lb.
(27) Percentage of ash and refuse to dry coal.........................per cent.
(28) Calorific value by calorimeter test per lb. of dry coal...............B.t.u.
(29) Cost of coal per ton of .... lb.......................................dollars.

Hourly Quantities.

(30) Dry coal consumed per hour, based on duration of running period.........lb.

Indicator Diagrams.

(31) Mean effective pressure in lb. per sq. in.................................lb.

Electrical Data.

(32) Total electrical output..kw.-hr.
(33) Electrical output per hour...kw.
(34) Output consumed by exciter...kw.
(35) Net electrical output per hour...kw.
(36) Average volts each phase..volts.
(37) Average amperes each phase..amperes.
(38) Power factor..

Speed.

(39) Revolutions per minute..rev.

Power.

(40) Indicated horse power developed by main engine:
First cylinder...i.h.p.
Second cylinder..i.h.p.
Whole engine...i.h.p.
(41) Net electrical horse power...h.p.

Economy Results.

(42) Dry coal consumed per i.h.p. per hour......................................lb.
(43) Dry coal consumed per kw.-hr...lb.
(44) Cost of coal per i.h.p. per hour..cents.
(45) Cost of coal per kw.-hr...cents.

* When an independent superheater is used, this includes coal burned in the superheater.

APPENDIX F.

RULES FOR CONDUCTING DUTY TRIALS OF STEAM PUMPING MACHINERY.*

A.S.M.E. Code of 1912.

1. OBJECT AND PREPARATIONS.

Read the general instructions given in Appendix A. Determine the object, take the dimensions, note the physical conditions not only of the pumping machinery but of all parts of the plant concerned, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

In a reciprocating pump, determine the quantity of water leakage or slip past the plungers, and that of the pump valves, if any.

2. APPARATUS AND INSTRUMENTS.

The apparatus and instruments required for a simple duty trial of pumping machinery, in which the steam consumption is determined by feed-water measurement, are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place).
(*b*) Graduated scales attached to the water glasses of the boilers.
(*c*) Pressure gauges, vacuum gauges, and thermometers.
(*d*) A steam calorimeter.
(*e*) A barometer.
(*f*) A tachometer or other speed-measuring apparatus.
(*g*) For rotary pumps a weir or other means for measuring the quantity of water pumped.
(*h*) Stroke scales, for direct-acting pumps.

In trials of a reciprocating pumping engine, involving the determination of the complete performance, the weir or other means of measurement noted should be provided, and in addition the following:

(*i*) Steam-engine indicators.
(*j*) A planimeter.

* Preliminary report of the Committee on Power Tests. (Jour. A.S.M.E., Nov., 1912.) In the case of a pump driven by some other prime mover than a steam engine or steam turbine, the code may be modified to suit the particular circumstances.

The determination of the heat and steam consumption should conform to the same methods as those described in the Steam Engine Code.

The steam consumed by steam-driven auxiliaries which are required in the operation of the pumping machinery should be included in the total steam from which the heat consumption is calculated, the same as noted in the Steam Engine Code.

3. OPERATING CONDITIONS.

Determine what the operating conditions should be to conform to the object in view and see that they prevail throughout the trial.

In trials for maximum duty, care should be taken that no air is snifted into the pump cylinders, causing imperfect filling. In such cases, and indeed in all cases where air is thus admitted in sufficient quantity to affect the performance as revealed by indicator diagrams from the water end, the result should be corrected accordingly.

4. DURATION.

5. STARTING AND STOPPING.

6. RECORDS.

The rules pertaining to the subjects Duration, Starting and Stopping and Records are identically the same as those given under the respective headings in the Steam Engine Code, and reference may be made to that code for the necessary directions in these particulars. Where the pump end is of the reciprocating class, the indicator diagrams should be taken not only from the steam cylinders but also from the water cylinders.

7. CALCULATION OF RESULTS.

The rules pertaining to Dry Steam, Heat Consumption, and Indicated Horse Power are identically the same as those given in the Steam Engine Code; and reference may be made to that code for the necessary directions in these particulars.

(*a*) *Water Horse Power.* The water horse power in a reciprocating pump is found by multiplying the net area of the plunger in square inches by the total head, which is made up of the pressure shown by the gauge on the force main, that on the suction main, and that representing the vertical distance between the centers of the two gauges, all expressed in pounds per square inch; the length of the stroke in feet; and the number of single strokes per minute; and dividing the final product by 33,000.

In a rotary pump the water horse power is found by multiplying the weight of water discharged per hour in pounds, as determined by weir or other

measurement; by the total head in feet, as determined from the readings of the gauge on the force main, that on the suction main, and the vertical distance between the two gauges; and the product divided by 1,980,000.*

(b) *Duty.* The duty per million heat units is found by dividing the number of foot-pounds of work done during the trial by the total number of heat units consumed; and multiplying the quotient by 1,000,000. The amount of work is found in the case of reciprocating pumps by multiplying the net area of the plunger in square inches, the total head expressed in pounds per square inches (which is made up of the pressure shown by the gauge on the force main, that on the suction main, and the vertical distance between the centers of the two gauges, all reduced to pounds), by the length of the stroke in feet, and the total number of single strokes during the trial; finally correcting for the percentage of leakage of the pump. In a rotary pump the work done is found by multiplying the weight of water discharged during the trial, as determined by weir or other measurement, by the total head in feet.

The duty per 1000 pounds of dry steam is found by dividing the foot-pounds of work done, as noted above, by the total weight of dry steam, and multiplying the quotient by 1000.

(c) *Capacity.* The capacity in gallons per 24 hours for reciprocating pumps is found by multiplying the net area of the plunger by the length of the stroke in feet (in direct-connected engines the average length of stroke); then by the number of single strokes per minute; and the product of these three by the constant 74.8; finally correcting for the percentage of leakage of the pump.

(d) *Leakage of Pump.* The percentage of leakage is the percentage borne by the quantity of leakage found on the leakage trial, to the quantity of water discharged on the duty run determined from plunger displacement.

(e) *Friction.* The percentage of total friction in a reciprocating pump is the percentage borne by the friction horse power to the indicated horse power of the steam cylinders.

(f) *Miscellaneous.* For the calculation of other results pertaining specially to the performance of the steam end of a reciprocating pump, reference may be made to the Steam Engine Code.

8. DATA AND RESULTS.

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

In the case of a pumping engine of the reciprocating class for which a record of the complete performance is desired, the additional engine data and results given in the Complete Form of the Steam Engine Code may supplement those here given.

* If there is a material difference in velocity of the water at the points where the gauges are attached, a correction should be made for the corresponding difference in "velocity head."

DATA AND RESULTS OF DUTY TRIAL OF STEAM PUMPING MACHINERY.

CODE OF 1912.

(1) Test of....................pump, located at..............................
to determine................, conducted by..............................
(2) Type of machinery..
(3) Rated capacity in gallons per 24 hours..................................
(4) Type of boiler...
(5) Type of auxiliaries..
(6) Dimensions of engine or turbine......................................
(7) Dimensions of pump...
(8) Dimensions of boilers..
(9) Dimensions of auxiliaries..
(10) Dimensions of condenser..
(11) Date..
(12) Duration..hr.

Average Pressures and Temperatures.

(13) Steam pressure at boiler by gauge..........................lb. per sq. in.
(14) Steam-pipe pressure near throttle, by gauge..................lb. per sq. in.
(15) Barometric pressure of atmosphere in inches of mercury................in.
(16) Pressure in receiver by gauge..............................lb. per sq. in.
(17) Vacuum in condenser in inches of mercury.............................in.
(18) Pressure in force main by gauge......................................lb.
(19) Pressure in suction main by gauge....................................lb.
(20) Vertical distance between centers of two gauges.......................ft.
(21) Total head, expressed in lb. pressure.................................lb.
(22) Total head, expressed in feet...ft.
(23) Temperature of main supply of feed water to boilers..................deg.
(24) Temperature of additional supplies of feed water.....................deg.
(25) Temperature of air in engine room....................................deg.

Total Quantities

(26) Water fed to boilers from main source of supply.......................lb.
(27) Water fed from additional supplies....................................lb.
(28) Total water fed to boilers from all sources...........................lb.
(29) Moisture in steam or superheating near throttle............. per cent or deg.
(30) Factor of correction for quality of steam, dry steam being unity.............
(31) Total dry steam consumed for all purposes.............................lb.
(32) Total leakage of plungers and valves..................................gal.
(33) Total number of gallons of water discharged:
(*a*) By plunger displacement, uncorrected..........................gal.
(*b*) By plunger displacement, corrected for leakage..................gal.
(*c*) By weir or other measurement of discharge.......................gal.
(34) Total weight of water discharged:
(*a*) By plunger displacement, corrected..............................lb.
(*b*) By measurement of discharge.....................................lb.

Hourly Quantities.

(35) Water fed from main source of supply . . . . lb.
(36) Water fed from additional supplies . . . . lb.
(37) Total water fed to boilers per hour . . . . lb.
(38) Total dry steam consumed per hour . . . . lb.
(39) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant . . . . lb.
(40) Net dry steam consumed per hour . . . . lb.
(41) Dry steam consumed per hour:
(a) By main engine or turbine . . . . lb.
(b) By auxiliaries . . . . lb.
(42) Water discharged per hour:
(a) By plunger displacement, corrected . . . . lb.
(b) By measurement of discharge . . . . lb.

Heat Data.

(43) Heat units per lb. of dry steam based on temperature, Line 23 . . . . . B.t.u.
(44) Heat units per lb. of dry steam based on temperature, Line 24 . . . . . B.t.u.
(45) Heat units consumed per hour based on main supply of feed . . . . . B.t.u.
(46) Heat units consumed per hour based on additional supplies of feed . . . . . B.t.u.
(47) Total heat units consumed per hour for all purposes . . . . . B.t.u.
(48) Loss of heat per hour due to leakage of plant, drips, etc . . . . . B.t.u.
(49) Net heat units consumed per hour . . . . . B.t.u.
(50) Heat units consumed per hour:
(a) By engine or turbine alone . . . . . B.t.u.
(b) By auxiliaries . . . . . B.t.u.

Indicator Diagrams.

(51) Mean effective pressure . . . . lb.

Speed and Stroke.

(52) Revolutions per minute . . . . rev.
(53) Number of single strokes per minute . . . . strokes.
(54) Average length of stroke . . . . ft.

Power.

(55) Indicated horse power developed:
1st cylinder . . . . i.h.p.
2d cylinder . . . . i.h.p.
Whole engine . . . . i.h.p.
(56) Water horse power . . . . h.p.
(57) Friction h.p. (Line 55 − Line 56) . . . . r.h.p.
(58) Percentage of i.h.p. lost in friction . . . . per cent.

Duty.

(59) Duty per million heat units . . . . ft.-lb.
(60) Duty per thousand lb. of steam . . . . ft.-lb.

Work Done per Heat Unit.

(61) Ft.-lb. of work per B.t.u. (Line 59 ÷ 1,000,000)........................ft.-lb.

Capacity.

(62) Number of gal. of water pumped in 24 hr.:
- (*a*) By plunger displacement, corrected for leakage....................gal.
- (*b*) By weir or other measurement....................................gal.

(63) Gallons of water per minute:
- (*a*) By plunger displacement, corrected for leakage....................gal.
- (*b*) By weir or other measurement....................................gal.

Economy Results, Steam End.

(64) Heat units consumed per i.h.p. per hour:*
- (*a*) By engine and auxiliaries..................................B.t.u.
- (*b*) By engine alone..B.t.u.
- (*c*) By auxiliaries...B.t.u.

(65) Dry steam consumed per i.h.p. per hour:
- (*a*) By engine and auxiliaries......................................lb.
- (*b*) By engine alone..lb.
- (*c*) By auxiliaries...lb.

* The i.h.p. on which the economy results are based is that of the main engine given in Line 54.

APPENDIX G.

RULES FOR CONDUCTING TESTS OF COMPLETE STEAM PUMPING MACHINERY PLANTS.*

A.S.M.E. Code of 1912.

1. OBJECT AND PREPARATIONS.

This code applies to a commercial test of a complete steam pumping machinery plant, having for an object the determination of the fuel cost of pumping a given quantity of water for a day's run of 24 hours. For tests of the component parts of the plant, rules may be found in the respective codes applying thereto.

Read the general instructions given in Appendix A. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out and prepare for the test accordingly.

In a reciprocating pump, ascertain the quantity of water leakage or slip past the plungers, and that of the pump valves.

2. FUEL.

Determine the character of the fuel to conform with the object in view. To obtain maximum economy or capacity, the fuel should be some kind of coal that is regarded as a standard, as noted in the Boiler Code.

3. APPARATUS AND INSTRUMENTS.

The apparatus and instruments required for a simple performance test of a complete pumping machinery plant are:

(*a*) Platform scales for weighing coal and ashes.
(*b*) A coal calorimeter.
(*c*) A tachometer or other speed-measuring apparatus.
(*d*) A pressure gauge attached to the force main.
(*e*) For rotary pumps, a weir or other means for measuring the quantity of water pumped.
(*f*) Stroke scales, for direct-acting pumps.

If the test also involves the determination of boiler performance and pumping machinery performance, additional apparatus and in-

* Preliminary report of the Committee on Power Tests. (Jour. A.S.M.E., Nov., 1912.)

struments are required, as pointed out in the respective codes referring thereto.

4. OPERATING CONDITIONS.

Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial.

5. DURATION.

The duration of a test of a complete pumping machinery plant should be not less than one day of 24 hours.

In cases where the machinery is in operation only a part of the calendar day, the coal consumption used in computing the results should be that of the entire 24 hours.

6. MISCELLANEOUS.

The methods of starting and stopping a test of a complete pumping plant, sampling and drying coal, determining ashes and refuse, and ascertaining the calorific value and chemical analysis of the coal, are the same as those described in the Code for Complete Steam Power Plants.

7. CALCULATION OF RESULTS.

For the calculation of water horse power, duty, capacity, and leakage of pump, reference may be made to the explanations given in the Pumping Engine Code.

8. RECORDS, DATA, AND RESULTS.

The records should be made as explained in Appendix A, and the tabular summary of data and results reported in accordance with the form given herewith, adding lines not provided for, or omitting those not required.

DATA AND RESULTS OF TEST OF STEAM PUMPING MACHINERY PLANT.

CODE OF 1912.

(1) Test of.................... plant located at..............................
to determine................ conducted by...........................
(2) Type of machinery...
(3) Rated capacity in gallons per 24 hrs.......................................
(4) Type of boilers...
(5) Type of auxiliaries..
(6) Dimensions of machinery...
(7) Dimensions of boilers...

(8) Dimensions of auxiliaries. . . . .
(9) Dimensions of condenser. . . . .
(10) Date. . . . .
(11) Duration. . . . . .hr.
(12) Length of time machinery was in motion with throttle open. . . . . .hr.
(13) Length of time machinery was running at normal speed. . . . . .hr.

AVERAGE PRESSURES AND TEMPERATURES.

(14) Steam pressure at boiler by gauge. . . . . .lb. per sq. in.
(15) Pressure in force main by gauge. . . . . .lb.
(16) Pressure in suction main by gauge. . . . . .in.
(17) Vertical distance between centers of two gauges. . . . . .ft.
(18) Total head expressed in lb. pressure. . . . . .lb.
(19) Total head expressed in feet. . . . . .ft.

SPEED AND STROKE.

(20) Revolutions per minute. . . . . .rev.
(21) Number of single strokes per minute. . . . . .strokes.
(22) Average length of stroke. . . . . .ft.

TOTAL QUANTITIES.

(23) Total coal as fired. . . . . .lb.
(24) Moisture in coal. . . . . .per cent.
(25) Total dry coal consumed. . . . . .lb.
(26) Ash and refuse. . . . . .lb.
(27) Percentage of ash and refuse to dry coal. . . . . .per cent.
(28) Calorific value per lb. of dry coal by calorimeter test. . . . . .B.t.u.
(29) Cost of coal per ton of ...... lb. . . . . .dollars.
(30) Total leakage of plungers and valves. . . . . .gal.
(31) Total number of gals. of water discharged:
(a) By plunger displacement, uncorrected. . . . . .gal.
(b) By plunger displacement, corrected for leakage. . . . . .gal.
(c) By weir or other measurement . . . . .gal.

HOURLY QUANTITIES.

(32) Dry coal consumed per hour, based on duration of running period (Line 25 ÷ Line 12). . . . . .lb.

POWER.

(33) Water horse power. . . . . .h.p.

ECONOMY RESULTS.

(34) Dry coal consumed per water h.p. per hour. . . . . .lb.
(35) Cost of coal per water h.p. per hour. . . . . .cents.
(36) Duty per 100 lb. of dry coal. . . . .

CAPACITY.

(37) Number of gal. of water pumped in 24 hr.:
(a) By plunger displacement, corrected for leakage. . . . . .gal.
(b) By weir or other measurement. . . . . .gal.
(38) Number of gal. of water pumped per minute:
(a) By plunger displacement, corrected for leakage. . . . . .gal.
(b) By weir or other measurement. . . . . .gal.

APPENDIX H.

RULES FOR CONDUCTING TESTS OF STEAM-DRIVEN COMPRESSORS, BLOWERS AND FANS.*

A.S.M.E. Code of 1912.

1. OBJECT AND PREPARATIONS.
2. APPARATUS AND INSTRUMENTS.
3. OPERATING CONDITIONS.
4. DURATION.
5. STARTING AND STOPPING.
6. RECORDS.

The directions pertaining to the above divisions of the subject are almost identical with those given in the Pumping Machinery Code under the same headings, the only difference being that the work done by the steam end is expended in moving a volume of air, instead of a body of water.

The quantity of air discharged should be measured by a gasometer, or by delivery into tanks of known capacity. Where these means are not available the pitot tube should be used, or some other means should be employed which is subject to calibration.

If the air end is of the reciprocating type, indicator diagrams should be regularly taken from this end as well as from the steam end.

7. CALCULATION OF RESULTS.

The rules pertaining to dry steam, heat consumption, and indicated horse power of the steam end, are identically the same as those given in the Steam Engine Code, and reference may be made to that code for the necessary directions in these particulars.

(*a*) *Air Horse Power.* The gross work done at the air end of a reciprocating machine, expressed in horse power, is found by multiplying together the net area of the air piston in square inches, the mean effective air pressure in pounds per square inch as determined from indicator diagrams, the length of

* Preliminary report of Committee on Power Tests. (Jour. A.S.M.E., Nov., 1912.) In the case of air machinery driven by some other prime mover than a steam engine or turbine, the code may be modified to meet the particular requirements.

the stroke in feet, and the number of single strokes per minute; and dividing their product by 33,000.

The net work at the air end of either reciprocating or rotary machines, expressed in foot-pounds per minute, is found by multiplying the corrected volume of the compressed air in cubic feet discharged into the main delivery pipe per minute, by the impact or total pressure in pounds per square feet and by the hyperbolic logarithm of the ratio of the total pressure to the atmospheric pressure (all pressures being absolute pressures). The net air horse power is found by dividing the product by 33,000. The corrected volume of the compressed air may be found by multiplying the sectional area of the delivery main in square feet by the mean velocity in feet per minute as determined by pitot tube or other measurement, and reducing the result to atmospheric temoerature by multiplying by the proportion

$$\frac{460 + t}{460 + T},$$

in which t is the temperature of the air supplied to the machine and T the temperature of the air in the delivery main.

(b) *Capacity.* The capacity is the number of cubic feet of air discharged through the delivery main per minute, as determined by gasometer, tank or other mode of measurement, reduced to the equivalent free air at the atmospheric temperature and pressure. The correction for pressure is made by multiplying by the proportion $\frac{P_2}{P_1}$, in which P_1 is the atmospheric pressure and P_2 the total pressure in the main (absolute pressures), and the correction for temperature as above.

The capacity may also be expressed in the number of cubic feet of compressed air discharged per minute at a given pressure above the atmosphere reduced to the atmospheric temperature.

(c) *Miscellaneous.* For methods of calculating results pertaining especially to the performance of the steam-end of a reciprocating air-pumping machine, reference may be made to the Steam Engine Code.

The "efficiency of compression" in a reciprocating machine is determined by first ascertaining the net work at the air end given above under the heading, "(a) Air Horse Power," and then dividing the net work thus found by the gross work given under the same heading.

The "mechanical efficiency" of a reciprocating machine is determined by dividing the gross air horse power at the air end by the indicated horse power at the steam end, or by the horse power delivered by the belt or motor in the case of other means of driving.

8. DATA AND RESULTS.

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for and omitting those not required, as may conform with the object in view.

In the case of an air-pumping machine of the reciprocating class for which a record of the complete performance is desired, the additional

engine data and results given in the Complete Form of the Steam Engine Code may supplement those here given:

DATA AND RESULTS OF TEST OF AIR MACHINERY. CODE OF 1912.

(1) Test of....................located at..................................
to determine...............conducted by....................
(2) Type of machinery..
(3) Rated capacity in cu. ft. of free air per minute.........................
(4) Rated capacity in cu. ft. of air discharged per minute at 100 lb. per sq. in. above atmosphere, reduced to the atmospheric temperature..........cu. ft.
(5) Type of boilers..
(6) Type of auxiliaries...
(7) Dimensions of engine or turbine at steam end.........................
(8) Dimensions of cylinders or blowers at air end.........................
(9) Dimensions of boilers.......................................
(10) Dimensions of auxiliaries.....................................
(11) Dimensions of condenser......................................
(12) Date...
(13) Duration..hr.

Average Pressures and Temperatures.

(14) Steam pressure at boiler by gauge..........................lb. per sq. in.
(15) Steam-pipe pressure near throttle, by gauge..................lb. per sq. in.
(16) Barometric pressure of atmosphere in in. of mercury.....................in.
(17) Pressure in receiver by gauge...............................lb. per sq. in.
(18) Vacuum in condenser in in. of mercury.............................in.
(19) Pressure in delivery main by gauge (impact pressure)..................lb.
(20) Total head, expressed in ft.......................................ft.
(21) Temperature of main supply of feed water to boilers...................deg.
(22) Temperature of additional supplies of feed water......................deg.
(23) Temperature of air in engine room or air supplied to machine...........deg.
(24) Temperature by wet-bulb thermometer..............................deg.
(25) Temperature of air in delivery main................................deg.

Total Quantities.

(26) Water fed to boilers from main source of supply.......................lb.
(27) Water fed from additional supplies...................................lb.
(28) Total water fed to boilers from all sources...........................lb.
(29) Moisture in steam or superheating near throttle.............per cent or deg.
(30) Factor of correction for quality of steam, dry steam being unity.............
(31) Total dry steam consumed for all purposes............................lb.
(32) Total cu. ft. of compressed air delivered as measured................cu. ft.
(33) Total cu. ft. of compressed air delivered reduced to atmospheric temperature and pressure...lb.
(34) Total weight of air delivered.......................................lb.

Hourly Quantities.

(35) Water fed from main source of supply . . . . lb.
(36) Water fed from additional supplies . . . . lb.
(37) Total water fed to boilers per hour . . . . lb.
(38) Total dry steam consumed per hour . . . . lb.
(39) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant . . . . lb.
(40) Net dry steam consumed per hour . . . . lb.
(41) Dry steam consumed per hour:
 (*a*) By main engine or turbine . . . . lb.
 (*b*) By auxiliaries . . . . lb.
(42) Cu. ft. of compressed air delivered per hour as measured . . . . cu. ft.
(43) Cu. ft. of compressed air delivered per hour reduced to atmospheric temperature . . . . cu. ft.
(44) Cu. ft. of compressed air delivered per hour reduced to atmospheric temperature and pressure . . . . cu. ft.
(45) Weight of air delivered per hour . . . . lb.

Heat Data.

(46) Heat units per lb. of dry steam based on temperature Line 21 . . . . B.t.u.
(47) Heat units per lb. of dry steam based on temperature Line 22 . . . . B.t.u.
(48) Heat units consumed per hour based on main supply of feed . . . . B.t.u.
(49) Heat units consumed per hour based on additional supplies of feed . . . . B.t.u.
(50) Total heat units consumed per hour for all purposes . . . . B.t.u.
(51) Loss of heat per hour due to leakage of plant, drips, etc . . . . B.t.u.
(52) Net heat units consumed per hour . . . . B.t.u.
(53) Heat units consumed per hour:
 (*a*) By engine or turbine alone . . . . B.t.u.
 (*b*) By auxiliaries . . . . B.t.u.

Indicator Diagrams.

(54) Mean effective pressure in steam cylinders . . . . lb.
(55) Mean effective pressure in air cylinders . . . . lb.

Speed and Stroke.

(56) Revolutions per minute . . . . rev.
(57) Number of single strokes per minute . . . . strokes.

Power.

(58) Indicated horse power developed at steam end of reciprocating machine . . . i.h.p.
(59) Gross air horse power as indicated in air cylinders of reciprocating machine . . . . air h.p.
(60) Net air horse power as computed from Line 43 . . . . air h.p.
(61) Friction of reciprocating machine (Line 58 − Line 59) . . . . fr.h.p.
(62) Percentage of i.h.p. lost in friction of machine . . . . per cent.

Economy Results, Steam End of Engine-driven Machines.

(63) Heat units consumed per i.h.p. per hour:
- (*a*) By engine and auxiliaries . . . . . B.t.u.
- (*b*) By engine alone . . . . . B.t.u.
- (*c*) By auxiliaries . . . . . B.t.u.

(64) Dry steam consumed per i.h.p. per hour: *
- (*a*) By engine and auxiliaries . . . . . lb.
- (*b*) By engine alone . . . . . lb.
- (*c*) By auxiliaries . . . . . lb.

Economy Results, Air Delivered.

(65) Heat units consumed per hour per net air h.p. of Line 60:
- (*a*) By engine or turbine and auxiliaries . . . . . B.t.u.
- (*b*) By engine or turbine alone . . . . . B.t.u.
- (*c*) By auxiliaries . . . . . B.t.u.

(66) Dry steam consumed per hour per net h.p. of Line 60:
- (*a*) By engine or turbine and auxiliaries . . . . . lb.
- (*b*) By engine or turbine alone . . . . . lb.
- (*c*) By auxiliaries . . . . . lb.

Efficiency Results.

(67) Thermal efficiency ratio for engine alone:
- (*a*) Per i.h.p., steam end (2545 ÷ Line 63*b*) . . . . . per cent.
- (*b*) Per net air h.p., air delivery (2545 ÷ Line 65*b*) . . . . . per cent.

Work Done per Heat Unit.

(68) Ft.-lb. of net work per B.t.u. consumed by engine or turbine and auxiliaries (1,980,000 ÷ Line 65*a*) . . . . . ft.-lb.

Capacity.

(69) Cu. ft. of compressed air delivered per minute as measured . . . . . cu. ft.

(70) Cu. ft. of compressed air delivered per minute, reduced to atmospheric temperature . . . . . cu. ft.

(71) Cu. ft. of compressed air delivered per minute at 100 lb. pressure, reduced to atmospheric temperature . . . . . cu. ft.

(72) Cu. ft. of compressed air delivered per minute, reduced to atmospheric temperature and pressure (free air) . . . . . cu. ft.

Miscellaneous Results.

Steam-driven Reciprocating Machine.

(73) Efficiency of compression (Line 60 ÷ Line 59 × 100) . . . . . per cent.

(74) Mechanical efficiency of machine (Line 59 ÷ Line 58 × 100) . . . . . per cent.

(75) Volumetric efficiency (Line 72 ÷ 1st Compr. Displ. × 100) . . . . . per cent.

* The i. h.p. on which these economy results are based is that of the main engine given in Line 58.

Note: In the case of air compressors having more than one stage and in those having intercoolers additional data should be given covering pressures and temperatures in the different stages, the quantity of water used for cooling and temperatures of the air and water entering and leaving the cooler.

APPENDIX I.

PROPERTIES OF SATURATED STEAM.* (Marks and Davis.)

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
P	t	q	r	$\lambda = r + q$	ρ	Apu	θ	$\frac{r}{T}$	$\theta + \frac{r}{T}$	S	γ
† 0.1	35.03	3.05	1071.7	1074.7	1017.3	54.4	0.0062	2.1666	2.1728	2935.0	0.000340
† 0.2	53.15	21.23	1061.6	1082.8	1005.2	56.5	0.0423	2.0704	2.1127	1524.0	0.000656
† 0.3	64.49	32.57	1055.3	1087.9	997.7	57.6	0.0640	2.0135	2.0775	1041.0	0.000961
0.4	72.91	40.95	1050.6	1091.6	992.4	58.5	0.0860	1.9730	2.0530	794.0	0.001259
0.5	79.68	47.71	1047.0	1094.6	987.6	59.3	0.0926	1.9413	2.0332	642.0	0.001555
0.6	85.32	53.34	1043.8	1097.1	983.9	59.9	0.1029	1.9155	2.0184	541.0	0.001850
0.7	90.18	58.18	1041.1	1099.3	980.7	60.4	0.1117	1.8936	2.0053	467.0	0.002143
0.8	94.46	62.45	1038.7	1101.2	977.8	61.0	0.1195	1.8747	1.9942	412.0	0.002431
0.9	98.33	66.31	1036.6	1102.9	975.2	61.4	0.1265	1.8578	1.9843	367.9	0.002719
1	101.83	69.8	1034.6	1104.4	972.9	61.7	0.1327	1.8427	1.9754	333.0	0.00300
2	126.15	94.0	1021.0	1115.0	956.7	64.3	0.1749	1.7431	1.9180	173.5	0.00576
3	141.52	109.4	1012.3	1121.6	946.4	65.8	0.2008	1.6840	1.8848	118.5	0.00845
4	153.01	120.9	1005.7	1126.5	938.6	67.0	0.2198	1.6416	1.8614	90.5	0.01107
5	162.28	130.1	1000.3	1130.5	932.4	68.0	0.2348	1.6084	1.8432	73.33	0.01364
6	170.06	137.9	995.8	1133.7	927.0	68.8	0.2571	1.5814	1.8285	71.89	0.01616
7	176.85	144.7	991.8	1136.5	922.4	69.4	0.2579	1.5582	1.8161	53.56	0.01867
8	182.86	150.8	988.2	1139.0	918.2	70.0	0.2673	1.5380	1.8053	47.27	0.02115
9	188.27	156.2	985.0	1141.1	914.4	70.6	0.2756	1.5202	1.7958	42.36	0.02361
10	193.22	161.1	982.0	1143.1	910.9	71.1	0.2832	1.5042	1.7874	38.38	0.02606
11	197.75	165.7	979.2	1144.9	907.8	71.5	0.2902	1.4895	1.7797	35.10	0.02849
12	201.96	169.9	976.6	1146.5	904.8	71.8	0.2967	1.4760	1.7727	32.36	0.03090

* Courtesy of the Publishers, Longmans, Green & Co. † Interpolated.

PROPERTIES OF SATURATED STEAM — (*Continued*).

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
p	t	q	r	$\lambda = r + q$	ρ	Apu	θ	$\frac{r}{T}$	$\theta + \frac{r}{T}$	S	γ
13	205.87	173.8	974.2	1148.0	902.0	72.2	0.3025	1.4639	1.7664	30.03	0.03330
14	209.55	177.5	971.9	1149.4	899.3	72.6	0.3081	1.4523	1.7604	28.02	0.03569
14.7	212.00	180.0	970.4	1150.4	897.6	72.9	0.3118	1.4447	1.7565	26.79	0.03732
15	213.0	181.0	969.7	1150.7	896.8	72.9	0.3133	1.4416	1.7549	26.27	0.03806
20	228.0	196.1	960.0	1156.2	885.8	74.3	0.3355	1.3965	1.7320	20.08	0.04980
25	240.1	208.4	952.0	1160.4	876.8	75.3	0.3532	1.3604	1.7136	16.30	0.0614
30	250.3	218.8	945.1	1163.9	869.0	76.2	0.3680	1.3311	1.6991	13.74	0.0728
35	259.3	227.9	938.9	1166.8	862.1	76.9	0.3868	1.3060	1.6868	11.89	0.0841
40	267.3	236.1	933.3	1169.4	855.9	77.6	0.3920	1.2841	1.6761	10.49	0.0953
45	274.5	243.4	928.2	1171.6	850.3	78.1	0.4021	1.2644	1.6665	9.39	0.1065
50	281.0	250.1	923.5	1173.6	845.0	78.6	0.4113	1.2468	1.6581	8.51	0.1175
55	287.1	256.3	919.0	1175.4	840.2	78.9	0.4196	1.2309	1.6505	7.78	0.1285
60	292.7	262.1	914.9	1177.0	835.6	79.7	0.4272	1.2160	1.6432	7.17	0.1394
65	298.0	265.7	911.0	1178.5	831.4	79.8	0.4344	1.2034	1.6368	6.65	0.1503
70	302.9	272.6	907.2	1179.8	827.3	80.1	0.4411	1.1896	1.6307	6.20	0.1612
75	307.6	277.4	903.7	1181.8	823.5	80.5	0.4474	1.1778	1.6252	5.81	0.1721
80	312.0	282.0	900.3	1182.3	819.8	80.7	0.4535	1.1665	1.6200	5.47	0.1829
85	316.3	286.3	897.1	1183.4	816.3	81.0	0.4590	1.1561	1.6151	5.16	0.1937
90	320.3	290.5	893.9	1184.4	813.0	81.2	0.4644	1.1461	1.6105	4.89	0.2044
95	324.0	294.5	890.9	1185.4	809.7	81.5	0.4694	1.1367	1.6061	4.65	0.2151
100	327.8	298.3	888.0	1186.3	806.6	81.7	0.4743	1.1277	1.6020	4.429	0.2258

105	331.4	302.0	885.2	1187.2	803.6	81.9	0.4789	1.1191	1.5980	4.230	0.2365
110	334.8	305.5	882.5	1188.0	800.7	82.1	0.4834	1.1108	1.5942	4.047	0.2472
115	338.1	309.0	879.8	1188.8	797.9	82.3	0.4877	1.1030	1.5907	3.880	0.2577
120	341.3	312.3	877.2	1189.6	795.2	82.5	0.4919	1.0954	1.5873	3.726	0.2683
125	344.4	315.5	874.7	1190.3	792.6	82.6	0.4959	1.0880	1.5839	3.583	0.2791
130	347.4	318.6	872.3	1191.0	790.0	82.8	0.4998	1.0809	1.5807	3.452	0.2897
135	350.3	321.7	869.9	1191.6	787.5	82.9	0.5035	1.0742	1.5777	3.331	0.3002
140	353.1	324.6	867.6	1192.2	785.0	83.0	0.5072	1.0675	1.5747	3.219	0.3107
145	355.8	327.4	865.4	1192.8	782.7	83.2	0.5107	1.0612	1.5719	3.112	0.3213
150	358.5	330.2	863.2	1193.4	780.4	83.3	0.5142	1.0550	1.5692	3.012	0.3320
155	361.0	332.9	861.0	1194.0	778.1	83.5	0.5175	1.0489	1.5664	2.920	0.3425
160	363.6	335.6	858.8	1194.5	775.8	83.6	0.5208	1.0431	1.5639	2.834	0.3529
165	366.0	338.2	856.8	1195.0	773.6	83.7	0.5239	1.0376	1.5615	2.753	0.3633
170	368.5	340.7	854.7	1195.4	771.5	83.8	0.5269	1.0321	1.5590	2.675	0.3738
175	370.8	343.2	852.7	1195.9	769.4	83.9	0.5299	1.0268	1.5567	2.602	0.3843
180	373.1	345.6	850.8	1196.4	767.4	84.0	0.5328	1.0215	1.5543	2.533	0.3948
185	375.4	348.0	848.8	1196.8	765.4	84.1	0.5356	1.0164	1.5520	2.468	0.4052
190	377.6	350.4	846.9	1197.3	763.4	84.2	0.5384	1.0114	1.5498	2.406	0.4157
195	379.8	352.7	845.0	1197.7	761.4	84.3	0.5410	1.0066	1.5476	2.346	0.4262
200	381.9	354.9	843.2	1198.1	759.5	84.4	0.5437	1.0019	1.5456	2.290	0.437
205	384.0	357.1	841.4	1198.5	757.6	84.5	0.5463	0.9973	1.5436	2.237	0.447
210	386.0	359.2	839.6	1198.8	755.8	84.5	0.5488	0.9928	1.5416	2.187	0.457
215	388.0	361.4	837.9	1199.2	754.0	84.6	0.5513	0.9885	1.5398	2.138	0.468
220	389.9	363.4	836.2	1199.6	752.3	84.7	0.5538	0.9841	1.5379	2.091	0.478
225	391.9	365.5	834.4	1199.9	750.5	84.7	0.5562	0.9799	1.5361	2.046	0.489
230	393.8	367.5	832.8	1200.2	748.8	84.8	0.5586	0.9758	1.5341	2.004	0.499
240	397.4	371.4	829.5	1200.9	745.4	85.0	0.5633	0.9676	1.5309	1.924	0.520
250	401.1	375.2	826.3	1201.5	742.0	85.1	0.5676	0.9600	1.5276	1.850	0.541
275	409.5	384.2	818.6	1202.8	734.2	85.3	0.5780	0.9419	1.5199	1.686	0.593
300	417.5	392.7	811.3	1204.1	726.8	85.6	0.5878	0.9251	1.5129	1.551	0.645

APPENDIX J.

EQUIVALENT VALUES OF ELECTRICAL AND MECHANICAL UNITS.

1 Myriawatt =

10 kilowatts
10,000 watts
13.41 horse power
13.597 cheval-vapeur
13.597 pferde-kraft
26,552,000 foot pounds per hour
8,605,000 gram calories per hour
3,670,000 kilogram meters per hour
34,150 B.t.u. per hour
1.02 boiler horse power

1 Kilowatt =

0.1 myriawatt
1,000 watts
1.341 horse power
1.3597 cheval-vapeur
1.3597 pferde-kraft
2,655,200 foot pounds per hour
860,500 gram calories per hour
367,000 kilogram meters per hour
3,415 B.t.u. per hour
0.102 boiler horse power

1 Horse Power =

745.7 watts
0.7457 kilowatt
0.07457 myriawatt
1.0139 cheval-vapeur
1.0139 pferde-kraft
33,000 foot pounds per minute
641,700 gram calories per hour
273,743 kilogram meters per hour
2,547 B.t.u. per hour

1 Cheval-Vapeur or Pferde-Kraft =

75 kilogram meters per second
0.07354 myriawatt
0.7357 kilowatt
0.9863 horse power
32,550 foot pounds per minute
632,900 gram calories per hour
2,512 B.t.u. per hour

1 Joule =

1 watt second
0.10197 kilogram meter
0.73756 foot pound
0.239 gram calorie
0.0009486 B.t.u.

1 Foot Pound =

1.3558 joules
0.13826 kilogram meter
0.001286 B.t.u.
0.03241 gram calorie
0.000000505 horse-power hour

1 B.t.u. =

1,054 watt seconds
777.5 foot pounds
107.5 kilogram meters
0.0003927 horse-power hour

1 Kilogram-Meter =

7.233 foot pounds
9.806 joules
2.344 gram calories
0.0093 kilogram meter

APPENDIX K.

MISCELLANEOUS CONVERSION FACTORS.

1 Pound per Square Inch =

2.0355 inches of mercury at 32° F.
2.0416 inches of mercury at 62° F.
2.309 feet of water at 62° F.
0.07031 kilogram per square centimeter
0.06804 atmosphere
51.7 millimeters of mercury at 32° F.

1 Foot of Water at 62° F. =

0.433 pound per square inch
62.355 pounds per square foot
0.883 inch of mercury at 62° F.
821.2 feet of air at 62° F. and barometer 29.92

1 Inch of Water 62° F. =

0.0361 pound per square inch
5.196 pounds per square foot
0.5776 ounce per square inch
0.0736 inch of mercury at 62° F.
68.44 feet of air at 62° F. and barometer 29.92

1 Foot of Air at 32° F. and Barometer 29.92 =

0.0761 pound per square foot
0.0146 inch of water at 62° F.

1 Inch of Mercury at 62° F. =

0.4912 pound per square inch
1.132 feet of water at 62° F.
13.58 inches of water at 62° F.

1 Atmosphere =

760.0 millimeters of mercury at 32° F.
14.7 pounds per square inch
29.921 inches of mercury at 32° F.
2,116.0 pounds per square foot
1.033 kilograms per square centimeter

1 Millimeter = 0.03937 inch

1 Centimeter = 0.3937 inch

1 Meter = 39.37 inches

1 Meter = 3.2808 feet

1 Square Meter = 10.764 square feet

1 Liter =

61.023 cubic inches
0.264 U. S. gallons

1 Gram =

1 cubic centimeter of distilled water
15.43 grains troy
0.0353 ounce

1 Kilogram =

2.20462 pounds avoirdupois

APPENDIX L.

TOTAL HEAT-ENTROPY DIAGRAM.

(MOLLIER DIAGRAM.)

The steam tables give values of the simultaneous physical properties of steam, such as pressure, entropy, temperature, etc. When certain of these properties are known the remainder can be obtained from the tables. The simultaneous properties can also be shown by means of a diagram each point on which represents steam in a perfectly definite condition.

Fig. 608 gives a skeleton outline of such a diagram and Fig. 609 a reduced reproduction of the complete chart as constructed by Marks

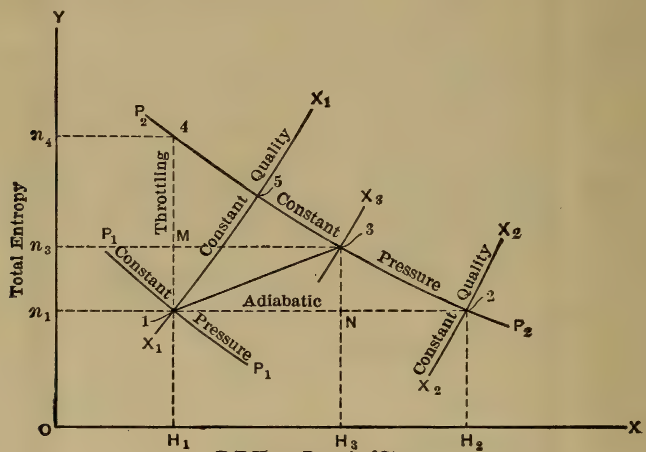
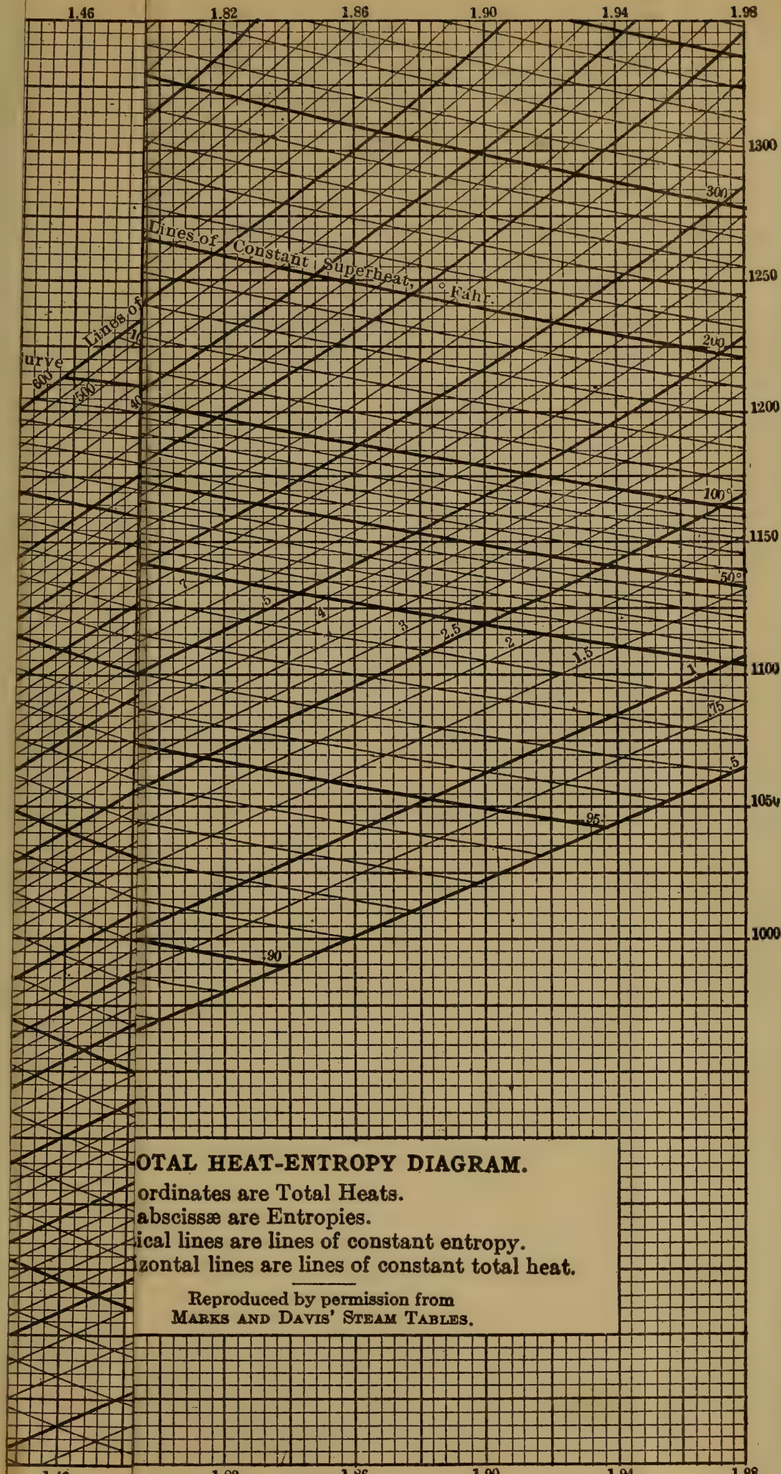


FIG. 608.

and Davis. Referring to Fig. 608, abscissas represent the heat contents or B.t.u. per pound of steam and ordinates represent the total entropy. Vertical lines then represent lines of constant heat content, and horizontal lines constant entropy. P_1P_1 and P_2P_2 represent lines of constant pressure and X_1X_1 and X_2X_2 lines of constant quality. Evidently any point in the chart represents a fixed condition of heat content, pressure, quality and entropy as determined by its location with respect to the different lines. Thus point 1 represents a pressure P_1 as



OTAL HEAT-ENTROPY DIAGRAM.

ordinates are Total Heats.

abscissæ are Entropies.

ical lines are lines of constant entropy.

izontal lines are lines of constant total heat.

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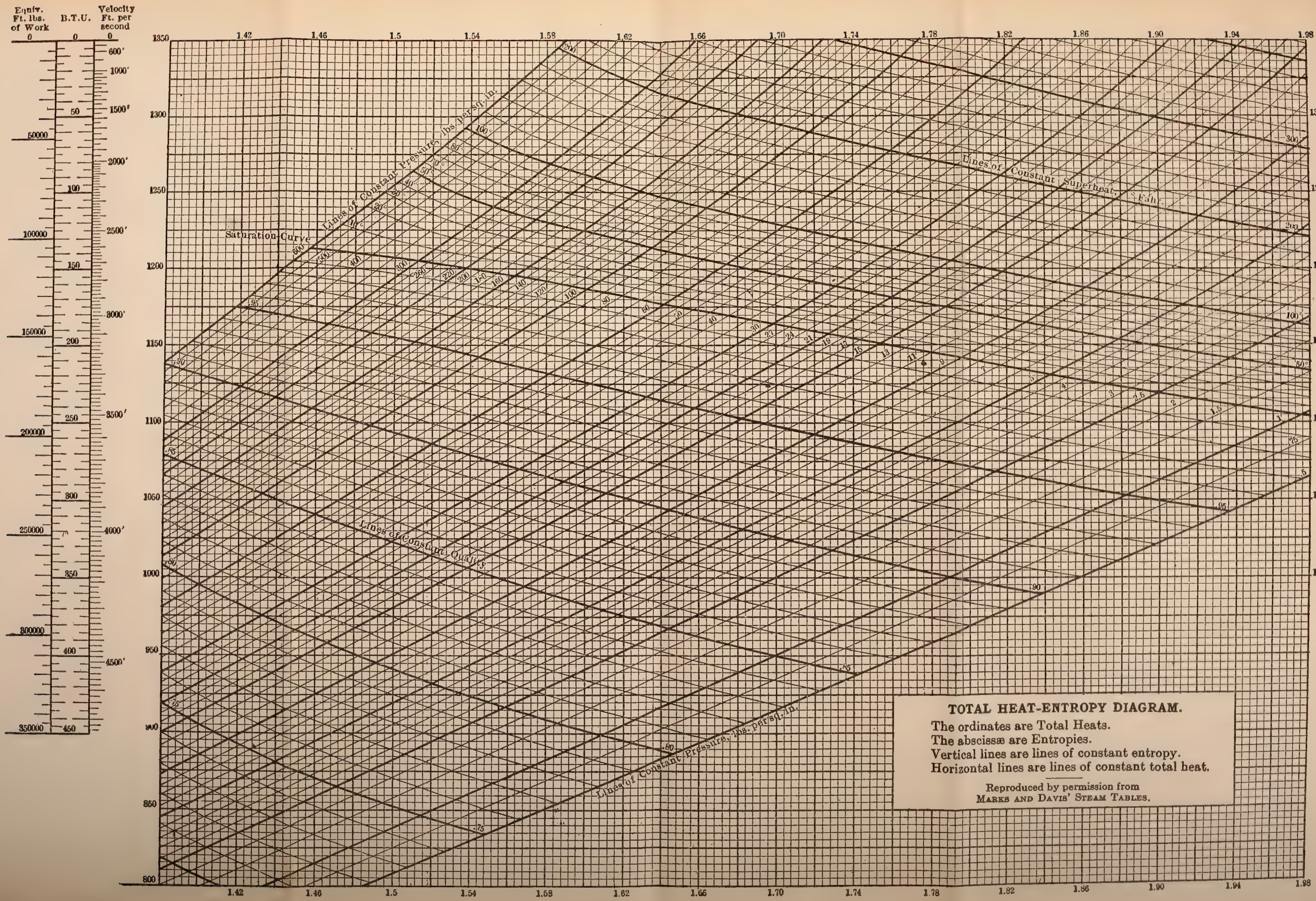

Equiv. Ft. lbs. of Work
B.T.U.
Velocity Ft. per second
Saturation Curve
Lines of Constant Pressure, lbs. per sq. in.
Lines of Constant Superheat, °Fahr.
Lines of Constant Quality
TOTAL HEAT-ENTROPY DIAGRAM.
The ordinates are Total Heats.
The abscissæ are Entropies.
Vertical lines are lines of constant entropy.
Horizontal lines are lines of constant total heat.
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FIG. 609.

determined by the numerical value of line P_1P_1, quality x_1 by its location on line X_1X_1, entropy n_1 by its projection N_1 on the Y axis, and heat content H_1 by its projection on the X axis.

The principal advantages of a total heat-entropy diagram over the tables are that they give the properties of wet and superheated steam and offer a simple means of solving many problems without calculations. For example, the chart offers a ready solution of problems involving

(*a*) Adiabatic expansion.
(*b*) Throttling.
(*c*) Expansion with frictional resistances.

(*a*) *Adiabatic Expansion:* From thermodynamics we know that during an adiabatic change the entropy is constant; thus, in expanding from pressure P_1 and condition represented in point 1 to a lower pressure P_2 it is only necessary to find the intersection 2 of a horizontal line from point 1 with line P_2P_2. The various properties corresponding to point 2 can be read directly from the diagram.

The line 1–2 $= H_1H_2$ represents the *difference* in heat content following adiabatic expansion from pressure P_1 and condition 1 to pressure P, or

line $$H_1H_2 = H_1 - H_2 = x_1r_1 + q_1 - x_2r_2 - q_2.$$

The quality x_2 is read directly from the intersection of line 1–2 with the constant quality line X_2X_2.

The entropy n_2, of course, remains the same.

From equation (129), p. 389, we find that the velocity due to adiabatic expansion is

$$V = 223.8 \sqrt{H_1 - H_2}.$$

Marks and Davis have added along the margin of the diagram (Fig. 609) a scale of velocity so that V may be ascertained by laying off the length H_1H_2 on the scale.

Example: Steam at 120 pounds absolute, quality 0.98, expands adiabatically to a back pressure of 2 pounds absolute. Find the quality and heat content at the lower pressure.

From Fig. 609 we locate P_1 at the intersection of pressure curve 120 and quality curve 0.98. The corresponding values of H_1 and n_1 are found by interpolation to be 1174.7 and 1.564 respectively. Follow vertical line 1.564 until it intersects pressure line "2". The corresponding values of H_2 and x_2 are found to be 910 and 0.797 respectively. The vertical intercept between the two pressure lines laid off on the velocity diagram gives V = 3640 feet per second.

Supposing the steam to be superheated 200 degrees instead of being wet, find the quality and heat content at the end of expansion.

Locate P_1 at the intersection of pressure curve 120 and superheat curve 200. The corresponding values of H_1 and n_1 are found to be 1295 and 1.703 respectively. Follow vertical line 1.703 until it intersects pressure line "2". The corresponding values of H_2 and x_2 are found to be 990 and 877 respectively.

(*b*) *Throttling:* If steam expands through a small orifice without the addition or abstraction of heat and is brought finally to its initial velocity its total heat will be unchanged. This process is called throttling and occurs when steam passes through a reducing valve. Vertical lines in Fig. 608 are lines of constant total heat and consequently show the changes in the condition of steam which result from throttling. Thus in throttling steam from pressure P_1, Fig. 608, to P_2 it is only necessary to find the intersection, 4, of a *vertical* line from point 1 with line P_2P_2.

Example: Steam at 200 pounds pressure and quality 0.96 passes through a reducing valve and its pressure is lowered to 15 pounds. Find its quality at the lower pressure. The intersection of pressure line 190, Fig. 609, with quality line 0.96 gives $H_1 = 1165$. Follow horizontal line 1165 until it intersects pressure line 15. The corresponding value for x_2 is found to be 30, that is, the steam is superheated 30 degrees.

To what pressure must the steam be reduced in order that it may be dry and saturated? Follow horizontal line 1165 until it intersects the *saturation* curve. The corresponding pressure is found to be 30 pounds.

(*c*) *Expansion Involving Frictional Resistances:* As steam expands in the nozzle of a turbine or passes between the vanes it experiences frictional resistances which cause it to give up less energy than it would under ideal conditions. The work of friction causes the entropy of the steam at its lowest temperature to be greater than it would be if adiabatic expansion occurred and serves to increase its dryness fraction.

If y one hundredths of the heat $H_1 - H_2$ (given up in adiabatic expansion) is lost due to friction, the heat available for useful work is

$$(1 - y)(H_1 - H_2),$$

the resulting velocity of the jet is

$$V = 223.8\sqrt{(1 - y)(H_1 - H_2)},$$

and the increase in quality of the exhaust steam is

$$\frac{y\,(H_1 - H_2)}{r_2}.$$

These equations may be readily solved by means of the diagram.

Referring to Fig. 608 line 1–3 represents an expansion from pressure P_1 to pressure P_2 with frictional resistances.

From the diagram

$$y = \frac{N\,2}{12} = \frac{H_3H_2}{H_2H_1}.$$

$$(1 - y)\,(H_1 - H_2) = \text{line } 1\,N = H_1H_3.$$

Increase in quality $= \dfrac{y\,(H_1 - H_2)}{r_2}$ = distance 2–3 between X_2X_2 and X_3X_3.

Increase in entropy $= N\,3 = N_1N_3$.

Example: Steam at 160 pounds absolute initial pressure, quality 0.97, expands through a nozzle to a back pressure of 2 pounds absolute. If 15 per cent of the heat energy is lost in friction, find the quality of the steam at the lower pressure and the velocity of the jet.

From Fig. 609 we locate P_1 at the intersection of pressure curve 160 and quality curve 0.97. The corresponding value of H_1 is 1170. Follow line 1170 vertically until it intersects pressure line "2". From the diagram we find for adiabatic expansion $H_2 = 910$ $x_2 = 0.797$. But the friction increases the heat content at the end of expansion an amount $0.15 \times H_1 - H_2 = 0.15\,(1170 - 910) = 39$, so that the final heat content $= 910 + 39 = 949$.

Follow pressure line "2" until it intersects heat line 949. The quality x_2 is found to be 0.836 and the entropy $n_2 = 1.632$. From the velocity scale we find $V = 3320$ feet per second for $H_1 - H_2 = (1170 - 949)$.

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